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PARAMETRIC VARIATIONS OF A HEAT BALANCED ENGINE.(U)

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# UNITED STATES NAVAL ACADEMY
## DIVISION OF
## ENGINEERING AND WEAPONS
## ANNAPOLIS, MARYLAND

PARAMETRIC VARIATIONS OF A
HEAT BALANCED ENGINE

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United States Naval Academy
Annapolis, Maryland

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Performance of a CFR engine over a wide range of experimental conditions is reported in detail for standard spark ignition operation and for heat balanced configurations. Operating conditions were mapped for primary combustion chamber and balancing chamber volumes giving nearly constant balancing ratios for three selected compression ratios; edge gap clearance was also varied. Three secondary air modes were investigated during optimization of performance giving more than a 30% increase in output over standard S.I.

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A new method of engine performance analysis, the Run Quality Index (RQI) is proposed to help evaluate the heat balanced engine and compare it to other engines.

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## II. INTRODUCTION AND STATE OF THE ART

The Heat Balanced Engine concept, first described in the open literature in Ref. 1, has been explored in considerable detail, both experimentally and analytically. A brief review of the overall effort is made here to acquaint the reader with the scope of investigation underway before delving into the current results of parametric variations on a CFR engine.

Transient heat flow analysis of the piston caps involved in this type engine (see Fig. IV-2, 3) has been performed[2,3] for several configurations and materials suggested by material analysis. The combination of material and heat transfer effects has caused failure of every design to date in its early development stages. The above studies were aimed at better understanding these specific interactions in order to reduce the number of design iterations in arriving at a stable geometry, the number to date averaging 3 to 4. The phenomenon of thermal etching has been observed[4] and is relevant to material selection. Proper geometric proportions as well as type of material are seen to be vital[2,3] to survival of the cap in the intense reaction zone between the combustion and balancing chamber.

To better understand the interactions leading to pressure exchange postulated earlier,[1] a preliminary non-steady gas dynamic analysis[5] was conducted with the conclusion that, even for the top dead center stationary piston boundary condition considered, pressure exchange was possible, with the attendant enhancement of combustion,from balancing chamber mass transport to the combustion front. Confirmation of the mass transport process (interface motion) has been obtained by direct color Fastex

photography, Fastex Schlieren and holographic interferometry in a single cylinder two-dimensional (square bore) glass walled engine.

While the current work reported on here for real engines was in process, an extension of the air standard cycle analysis[7] was made to fuel-air composition.[8] The results are discussed in more detail later.

The object of this experimental effort was to observe the effect of varying one independent experimental variable at a time, all others remaining constant. The method of determining parameters is described in the next section. The operating modes selected were based on past experience[1], were influenced by the "learning" curve of a new investigator (CCF), and had the additional aim of optimizing performance based on geometric proportions interpreted from the results of air-standard[7] and fuel-air analysis.[8] While it was recognized (and emphasized) that the above idealized thermodynamic models establish only the theoretical limits attainable, certain trends predicted were, in fact, observed experimentally.

## III. DETERMINATION OF PARAMETRIC VARIATIONS

Prior to this work, experiments on the heat balanced engine had been conducted to establish overall limits of its capability.[1] Due to time limitations and monetary constraints experiments did not include the monitoring of all variables and in some cases more than one variable was changed between runs. Air/fuel ratio, in particular, was not measured. As a result, it was difficult to interpret the performance data in the classical sense.

Previous experience had also shown that the number of variables for testing the heat balanced engine is greater than for other engines. There are standardized procedures for testing OTTO and DIESEL cycle engines that are accepted by industry. However, the NAHBE is not an OTTO or DIESEL cycle engine and standardized test procedures do not allow examination of all of the variables.

For this test series, it was decided initially that each of the variables would be varied independently. Previous results indicated that the optimum balancing ratio was about 0.5 and the optimum pressure exchange cap clearance was about .080" for heat balanced modification of the CFR engine. However, the interrelations between compression ratio, fuel grade, air/fuel ratio, engine speed, and load, with the best balancing ratio and cap clearance were unclear. Theory[6] indicates there is a synergistic effect, furthermore, the system is not linear. For-instance the maximum output changes with compression ratio, with balancing ratio, and with most of the other variables. It was decided then, that each variable would be changed through three to five values while all other variables were held constant. There are well defined experimental

procedures whereby the number of changes in variables can be significantly reduced, but to be absolutely sure of the effect of each variable, it was decided that no abreviation in the series would be made. However, this constraint was relaxed slightly as will be discussed later.

Variables considered and used were as follows:

Fuel: Since this is a spark ignition CFR engine, commercial grade gasoline was used. Exxon regular (89.9 approx. octane) was used as the standard. The highest commercial octane available (approx 96.9) was Sunoco 260. No suitable low octane gas was available therefore scientific heptane was mixed with Exxon regular in order to provide a low octane gas of 87.9 octane (see Table III-1). Sufficient fuel to complete all but the Mode 3 series was purchased at one time to preclude variation in quality.

Engine speed: The CFR engine runs well from 900 to 1500 RPM. Three speeds were used; 1100, 1300, and 1500 RPM.

Spark: An ignition timing series was planned; once run,an optimum spark setting would be selected and this parameter fixed for all runs.

Air-fuel ratio: Initially a single secondary air setting was used for all runs. Fuel flow was changed by altering the height of the fuel column at the fuel jet. Five different heights were used. Because of the ease in changing this variable, 5 settings were used instead of 3 as per other variables. Engine performance indicated other secondary air settings should be used. After the first series of runs were completed (Mode 1), the air intake system was modified as described later and other series were completed.

Gap: The distance between cylinder wall and the outer radius of the pressure exchange cap was changed by machining the cap. Three clearances

TABLE III-1

Test Fuel Octane Ratings

SUNOCO 260 (Octane)

| | |
|---|---|
| Research | 100.9 |
| Motor | 92.9 |
| Combine R+M | 96.9 |

EXXON REG. (Octane)

| | |
|---|---|
| Research | 94.0 |
| Motor | 85.7 |
| Combine R+M | 89.7 |

EXXON LOW (Octane)*

| | |
|---|---|
| Research | 91.2 |
| Motor | 84.5 |
| Combine R+M | 87.9 |

*Low Octane Mix: 25 gals Esso Regular, 1 gal Heptane

were tested; .070", .080" and .090".

Balancing Ratio: The balancing ratio is defined[1] as the ratio of the volume below the pressure exchange cap to the volume above the cap at TDC.

$$\beta = \frac{V_B}{V_A}$$

This ratio was varied experimentally by inserting spacers between the piston face and the pressure exchange cap (see Fig. III-1).

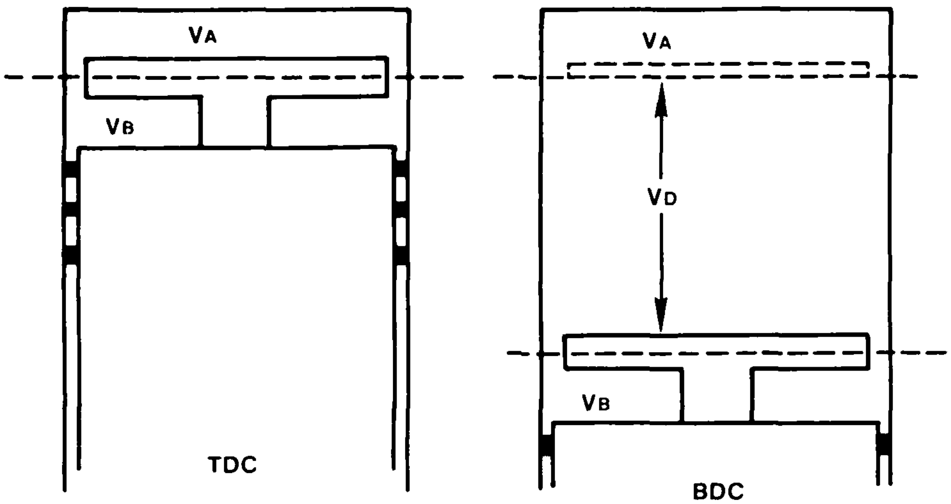
Compression Ratio: The compression ratio is defined as the total volume at BDC compared to the total volume at TDC. The compression

$$r = \frac{V_D + V_C}{V_C} \; ; \quad V_C = V_A + V_B$$

ratio was changed by varying the height of the cylinder head of the CFR engine (standard procedure, new calibration).

A problem arises since changing the compression ratio alters the CFR balancing ratio and vice versa. To maintain constant balancing ratio for varying compression ratio, a different spacer must be inserted for each compression ratio. Since this would have been very time consuming, it was more economical to let the balancing ratio vary a little as the compression ratio was changed. This gave an acceptable range of values for r and $\beta$.

To determine the required geometries, a computer program was developed to calculate the volume under the cap ($V_B$) as a function of $\beta$ for various values of r. The fifteen most suitable combinations of $\beta$ and r were chosen for five values of $V_B$ (see Fig. III-2). Four spacers were fabricated accordingly. This scheme reduced by three the number of times the engine had to be disassembled.




VA = VOLUME ABOVE CAP (COMBUSTION CHAMBER) @ TDC

VB = VOLUME BELOW CAP (BALANCING CHAMBER)

VC = VA + VB, CLEARANCE VOLUME

VD = DISPLACEMENT

r = ( VD + VC) /VC, COMPRESSION RATIO

**Figure III-1**
**General Terminology for Heat Balanced Engine**

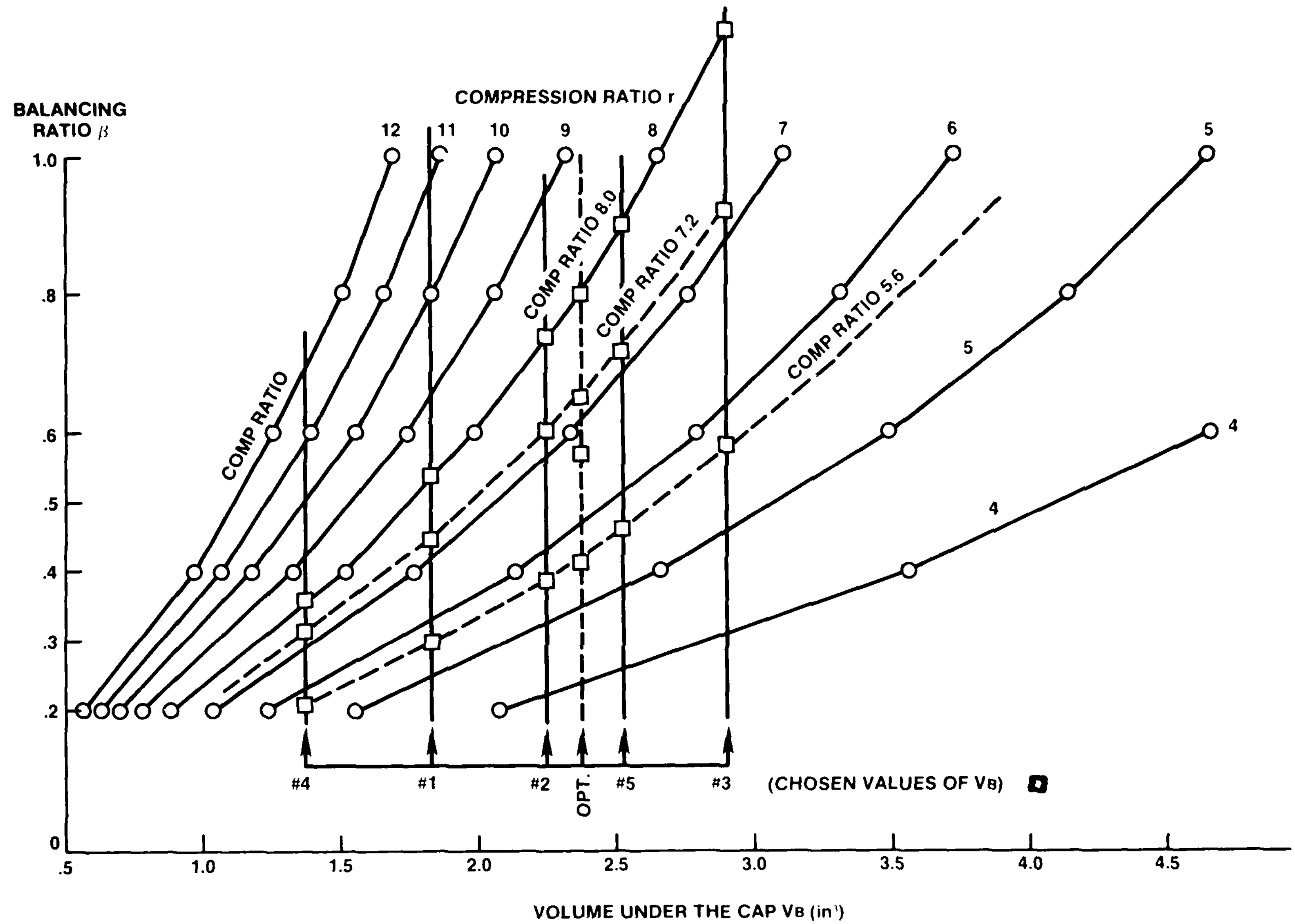



Figure III-2
Method of Selecting $V_B$ for Varying Compression Ratio, Balancing Ratio

An estimate of the number of runs could now be made. Defining the number of independent variables we have:

| OCTANE | | RPM | | FUEL HT | | GAP | | COMP. RATIO | | BAL. RATIO | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3 | x | 3 | x | 5 | x | 3 | x | 3 | x | 3 | = | 1215 runs. |

This number was significantly reduced by establishing a clearance gap early in the test series and noting that the engine performed well on all three fuels. Although the result was a long tedious process, synergistic effects were minimized and the data provided conclusive information concerning engine performance.

# IV. EXPERIMENTAL APPARATUS

## Engine

The Co-operative Fuel Research engine (ASTM-CFR) utilized in this test series is specifically designed as a precision engine having independent control over compression ratio, ignition timing, air-fuel ratio and load while operating within a range of 900-1500 rpm. Modifications to the basic CFR engine fall into two categories: piston design and carburetor fuel/air system.

PISTON: A standard CFR steel piston was modified to accommodate a series of steel pressure exchange caps. These caps (Fig. IV-1) utilized a common design and varied principally in edge gap (.070, .080 and .090 inches) and volume under the cap, ($1.35 \leq V_B \leq 2.90$ in$^3$). Volume was varied through the use of steel spacers (Fig. IV-1). Both cap and spacer were attached to the piston with a 1/2 inch diameter steel bolt (Fig. IV-2).

CARBURETOR FUEL/AIR SYSTEM: Three primary changes were made to the standard carburetion system. First, each fuel (low, medium and high octane) was fed through one of three adjustable fuel bowls and into a single fuel metering jet (Fig. IV-3). This individuality was necessary to insure fuel homogeneity and consistency in metering. Next, the main carburetor air inlet was fitted with a choke and plenum chamber. This device acted essentially as a means of varying the air/fuel ratio and allowed fine tuning of primary air into the engine.

The last and perhaps most important modification was the secondary air system. This consisted of a 3/4 inch internal diameter air inlet tube, having a gate valve on one end and a union coupling on the other. See Fig. IV-4.

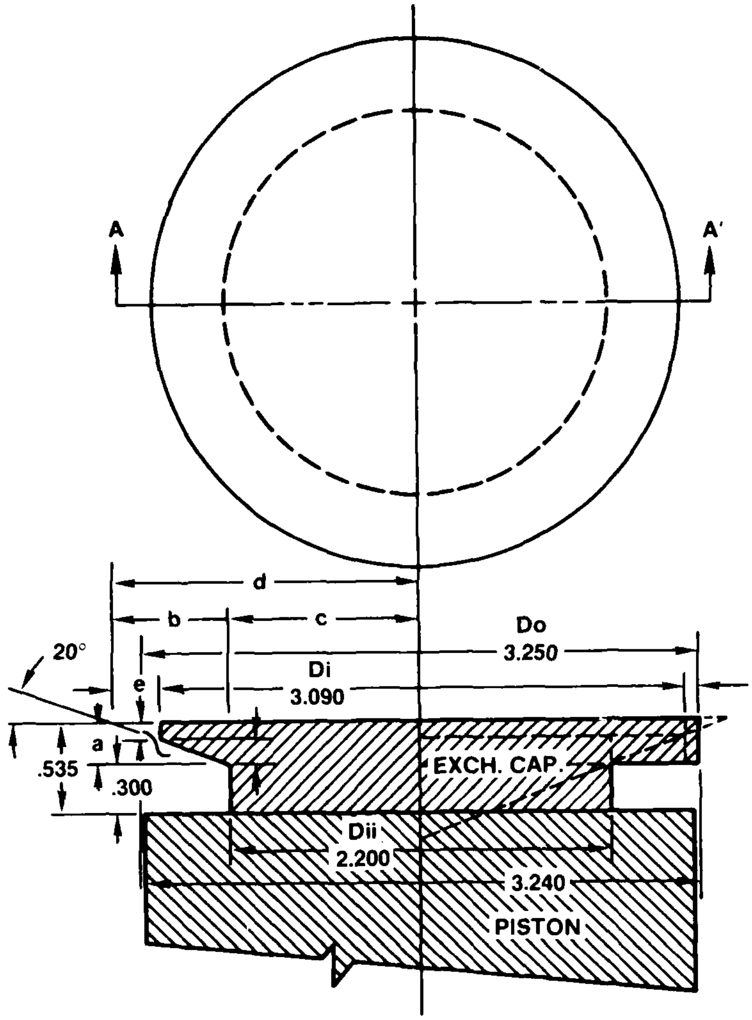



Fig. IV-1
Basic Pressure Exchange Cap Design

Fig. IV-2
Piston Assembly

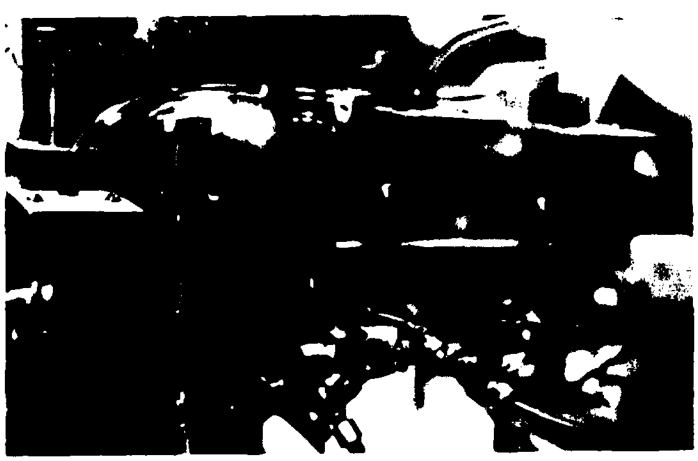


Fig. IV-3
Carburetor Assembly

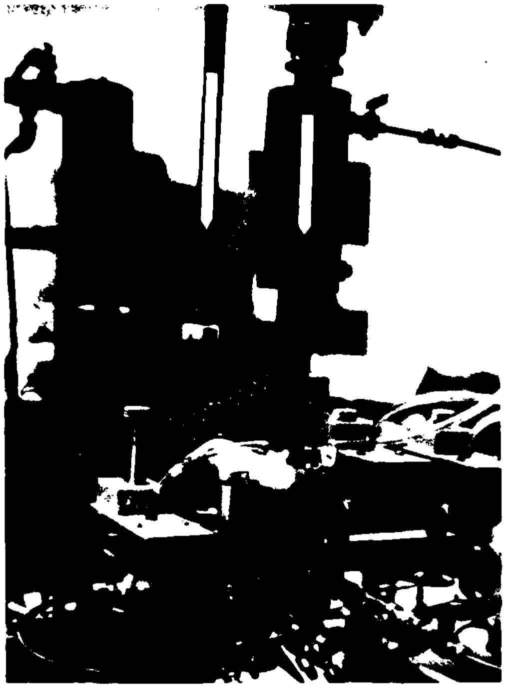


Fig. IV-4
Primary & Secondary Inlets

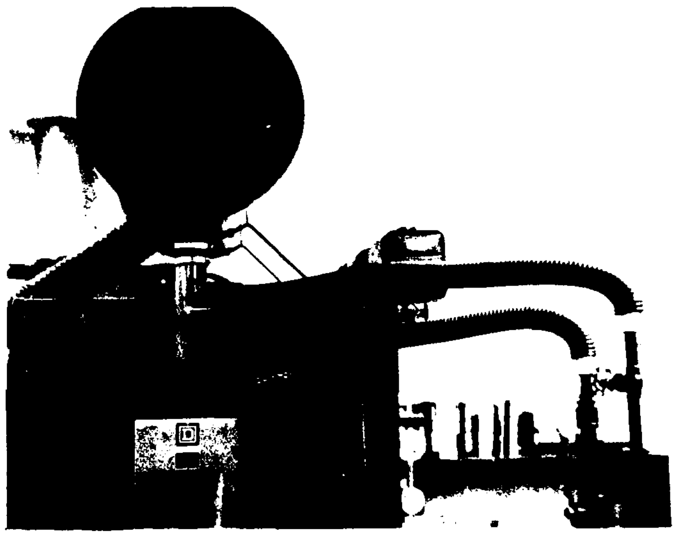


Fig. IV-5
Air Inlet Plenum With Rubber
Pulsation Damper Shown

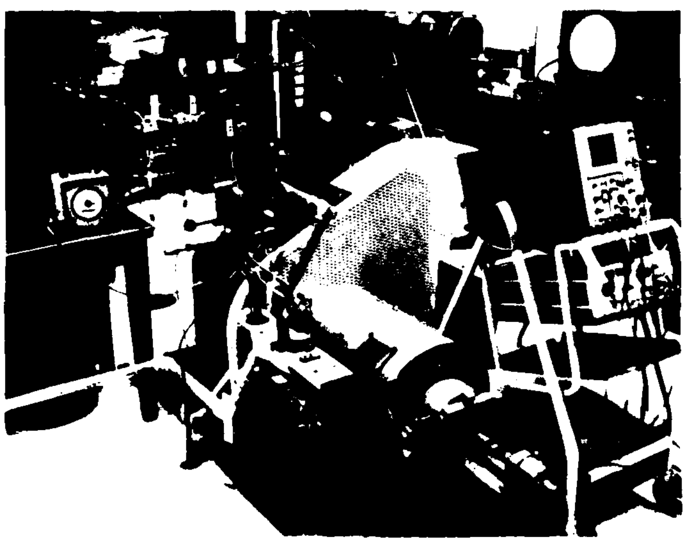


Figure IV-6
Test Bed Apparatus

It was installed between the carburetor venturi and the engine intake manifold. In Mode 2 described later (Section VI), a copper diaphragm having a .106 in diameter hole in its center was placed inside the lower union coupling. This diaphragm metered the bleed or secondary air into the engine. The secondary air system as a whole provided a means of ingesting air into the combustion chamber prior to the rich air-fuel mixture entering from the venturi.

Engine Performance Monitoring Devices

The CFR engine was instrumented to monitor the following parameters:

AIR: All airflow into the engine entered through a large plenum (Fig. IV-5). Pressure differential was measured to the nearest .5 mm $H_2O$ using a vacuum gauge and inclined manometer, Fig. IV-6.

FUEL: Three different octane gasoline fuels were tested (Table III-1). Each fuel was routed through a fuel control panel into a separate fuel bowl having an adjustable fuel level. A 50 cc burette was used to measure fuel quantities; run time and total crankshaft revolutions were recorded for each 50 cubic centermeters of fuel burned.

EMISSIONS: A Beckmen Model 590 Exhaust Gas Analyzer measured percent carbon monoxide and unburnt hydrocarbons (ppm hexane).

TEMPERATURE: Temperatures were recorded for six engine locations using type K Chromel-Alumel Thermocouples linked to a series 250 Omega Digital Temperature Indicator. The six measuring points were:

- Inlet air
- Intake manifold
- Fuel
- Cylinder head
- Water jacket
- Exhaust manifold EGT

PRESSURE: Combustion chamber head pressures were obtained using an air cooled PCB Piezoelectric spark plug transducer intergrated with a Tektronic Rotational Function Generator and a standard oscilloscope. This combination provided both pressure crankshaft angle and pressure-volume traces.

An overall view of the test bed apparatus is given in Fig. IV-6.

## V. OPERATING MODES, STANDARD SPARK IGNITION

Evaluation of the performance of the standard spark ignition engine was made for a quantative comparison with the Heat Balanced engine. A standard piston was used and the engine run at full open throttle. Test variables were set as follows:

- Compression ratio - 8.0:1, 7.2:1, 5.6:1
- rpm - 1100, 1300, 1500
- Fuel - high, medium and low octane (Table III-1)
- Spark - 15° BTDC

The only independent variable was that of air/fuel ratio which varied with the level of the carburetor fuel bowl assembly.

It was noted that smooth engine performance was difficult or impossible to maintain at fuel levels lower than 3.6 inches below the venturi centerline with a .033" diameter fuel jet. Therefore, levels of 2.0, 2.4, 2.8, 3.2 and 3.6 inches were used. This trend of unstable lean operations was experienced only with the standard spark ignition engine and was not a problem during runs using a pressure exchange cap and secondary air of the heat balance configuration.

## VI. OPERATING MODES, HEAT BALANCED

### Mode 1: Fixed Geometry Secondary and Primary Air

Mode 1 is the result of an independent interpretation (CCF) of the results previously reported.[1] The separate modes, 1, 2, and 3 can be viewed as the typical "learning curve" that any investigator might experience in transitioning from classical Otto operation to Heat Balanced Operation. This particular operating mode used a fixed secondary air orifice 0.143 inches in diameter at the manifold end of the secondary air inlet (Fig. VI-1) and unobstructed primary air at the carburetor (open throttle). No adjustment was made to the airflow through the primary or secondary inlets under operating conditions.

Variable air/fuel ratios were obtained by changing the level of the carburetor fuel bowl assembly. Fuel heights of 2.0, 3.0, 4.0, 4.5, and 5.0 inches below the venturi center line were used. Levels of approximately two inches gave a rich mixture and levels of five inches yielded very lean operation. Tests using high, mid, and low octane fuels of table III-1 were performed in this mode.

Pressure exchange cap design used was similar to that of Ref. (1) with the exception that the secondary volume under the cap ($V_B$) was varied by inserting spacers under the cap. This allowed variation of the volume for fixed edge gap clearance. Conversely, for fixed $V_B$, caps were fabricated with varying clearance gap. Thus the two parameters Beta ($\beta$), the ratio of the volume below the cap divided by the volume above the cap at top dead center ($\beta = \frac{V_B}{V_A}$), and edge gap clearance (g),

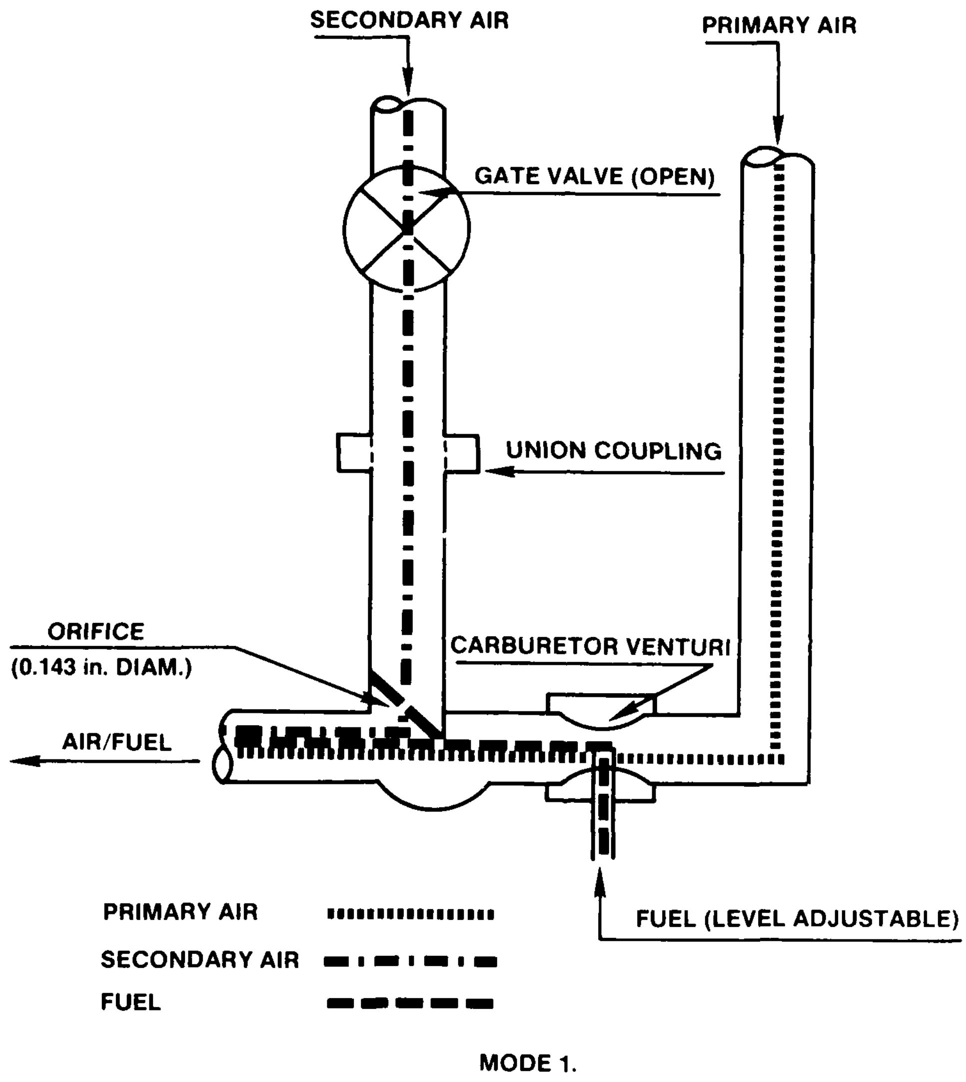



**Figure VI-1**
**Schematic of Primary and Secondary Air Control**

the distance between the cap's extreme radius and the cylinder wall were varied independently. The first cap was designed at a Beta of approximately 0.50 for a compression ratio of 8.0:1. Careful volumetric measuring with the cap installed provided a precise value of Beta equal to 0.54 at this compression ratio. Beta for other compression ratios was determined analytically (see Section III).

Previous performance[1] suggests an edge gap clearance of 0.080 inches as optimum. Therefore, caps having edge gaps of 0.070, 0.080, and 0.090 inches were used to verify earlier results. It is suspected that an excessive gap allows the fuel-air mixture to enter the balancing chamber, hereafter referred to as "poisioning" and causes ignition below the cap. On the other hand too small a gap hinders both filling and purging of gases in the balancing chamber and restricts wave interaction between the two chambers. This wave phenomenon is intimately related to the interface (between burned and unburned gas) that ultimately prolongs the combustion process in the heat balanced cycle.

A preliminary test series, run at a compression ratio of 7.0:1, 1100 rpm, on regular gas at a fuel level of 2.5 inches below the venturi was conducted to optimize and fix the ignition timing variable. Results indicated that setting of fifteen degrees before top dead center (15° BTDC), provided best all around performance; all subsequent testing was with this timing.

Internal friction load was considered next. Careful measurements were required to construct plots Fig. (VI-2) of frictional torque vs. RPM for various compression ratios. To perform these tests accurately the engine had to be stabilized at operating temperature and the crackcase oil temperature maintained at 105± 2°F. It is important to note that the friction torque

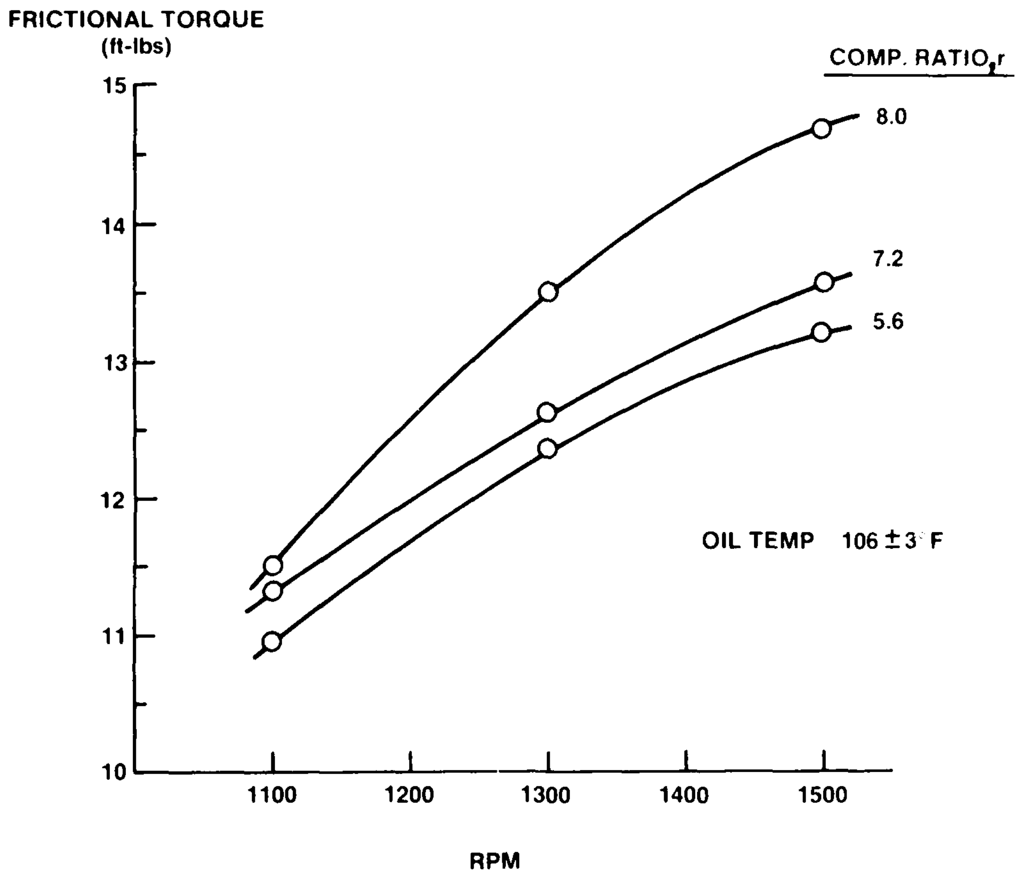



**Figure VI-2**
**Frictional Torque at Selected Compression Ratios**

required to motor the CFR engine is considerably higher than that of a normal internal combustion engine since this engine is overdesigned using heavy components and a special damping system for fuel research applications (ASTM standard).

Frequent checks on frictional torque were made to insure no shifts in operating conditions had occurred during any mode discussed here.

Mode 2: Fixed Geometry Secondary and Variable Primary Air

Marginal control over air/fuel ratio experienced with the air/fuel design used in operating Mode 1 suggested a redesign of both the primary and secondary air inlets and a change in testing technique. Redesign allowed the engine to be stabilized at a substantially greater range of air/fuel ratio and this was designated as operating Mode 2.

The secondary air metering orifice was the first change made. Mode 1 testing suggested that the amount of secondary air entering the engine was too small and either orifice size and/or diaphragm location was responsible. Therefore, the orifice, at the extreme end of the manifold was removed and replaced by an easy to change, flat copper diaphragm located inside the union coupling just above the intake manifold (fig. VI-3). This change in location and design simplified the orifice size optimization tests that followed.

The relocation of the orifice above the manifold also provided an air accumulator chamber between the intake manifold and the diaphragm that continuously charged itself with air throughout the engine cycle and vented to the combustion chamber only during the intake stroke. As might be expected because of the accumulator, the results of a series of tests attempting to optimize this parameter suggested an orifice size slightly smaller than the Mode 1 design, 0.106 inches vice .143 inch diameter.

Next, the primary air inlet chamber was modified to incorporate a choke and fine tuning valve. This new assembly enabled the engine to be run with a higher intake manifold vacuum and provided a means of accurately varying the air/fuel ratio while maintaining a fixed fuel level.

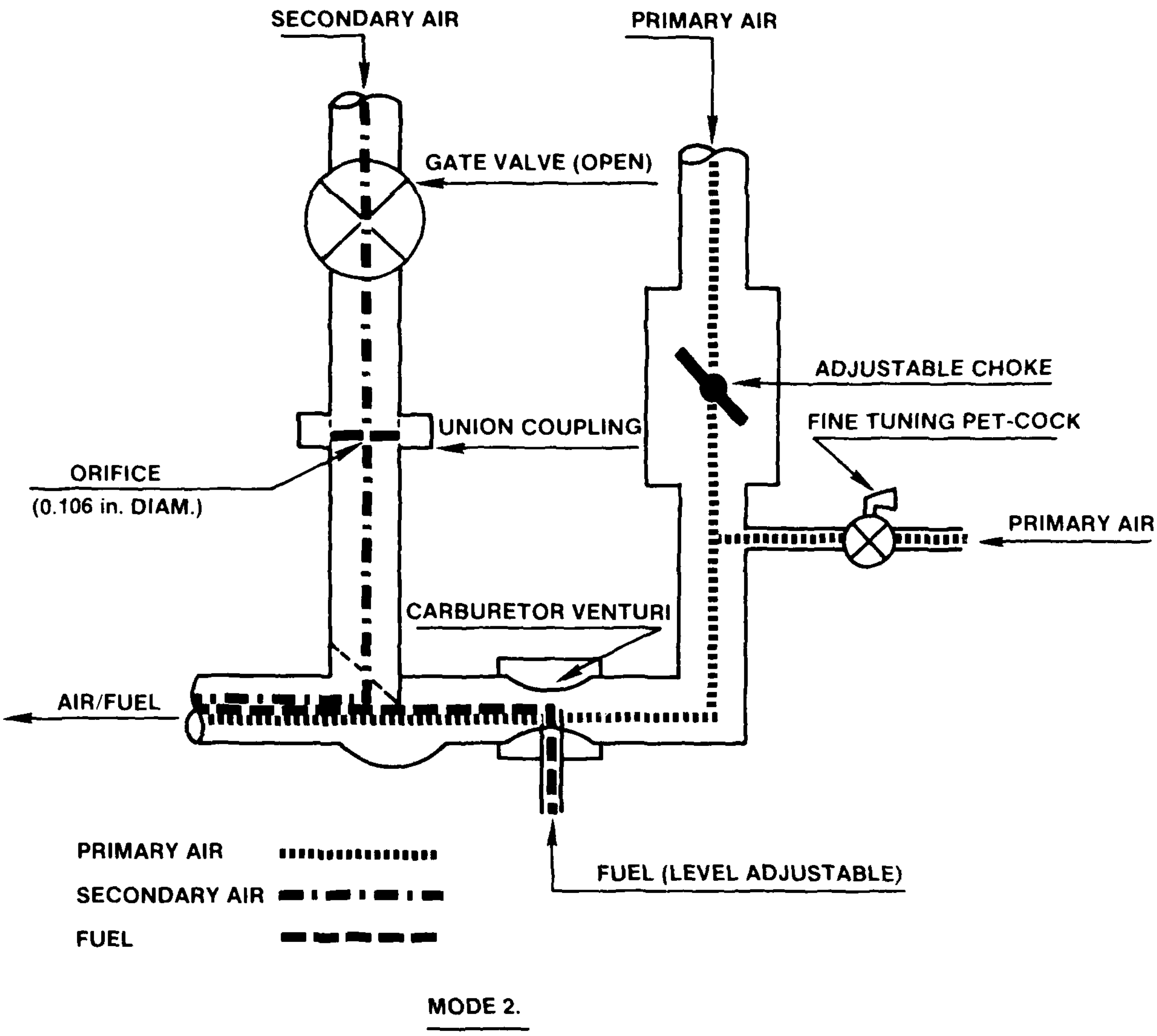



**Figure VI-3**
**Schematic of Primary and Secondary Air Control**

A Mode 2 operating procedure was established. Testing commenced at a fuel level of 2.0 inches, no choke, and at a fixed rpm. Next, fuel height was systematically lowered to 3.0, 4.0, 4.5, and 5.0 inches below the venturi. Also, the primary air inlet was gradually choked at each decrease in fuel height in order to establish leaner and leaner (overall composition) stable operating points. The fine tuning valve was used to extract peak engine performance at each setting.

Pressure exchange cap #1A (Table VI-1) was tested once again but this time following Mode 2 operating procedures. The engine performed in the heat balanced mode demonstrating stable runs over a large range of air/fuel ratios. Subsequent caps having edge gap clearances of .070 and .090 inches and the same $V_B$ were tested in a similar manner. Results suggested an optimun edge gap of .080 inches and this parameter was fixed for the remaining series. Caps #2 through #5 (table VI-1) were designed, fabricated and testing using the Mode 2 modification. This series of parametric tests provided the optimum $\beta$ as a function of compression ratio and RPM.

TABLE VI-1

| Cap NO | r | β | t" (spacer) | g" |
|---|---|---|---|---|
| #1A | 5.6 | .30 | None | .080 |
| | 7.2 | .44 | " | " |
| | 8.0 | .54 | " | " |
| 1B | 5.6 | .30 | None | .070 |
| | 7.2 | .44 | " | " |
| | 8.0 | .54 | " | " |
| 1C | 5.6 | .30 | None | .090 |
| | 7.2 | .44 | " | " |
| | 8.0 | .54 | " | " |
| 2 | 5.6 | .38 | +0.100 | .080 |
| | 7.2 | .60 | " | |
| | 8.0 | .74 | " | |
| 3 | 5.6 | .58 | +0.239 | .080 |
| | 7.2 | .91 | | |
| | 8.0 | 1.18 | | |
| 4 | 5.6 | .21 | -0.100 | .080 |
| | 7.2 | .31 | " | " |
| | 8.0 | .36 | " | " |
| 5 | 5.6 | .46 | +0.157 | .080 |
| | 7.2 | .71 | " | " |
| | 8.0 | .90 | " | " |
| OPT | 5.6 | .41 | +0.120 | .080 |
| | 6.6 | .57 | " | " |
| | 7.2 | .64 | " | " |
| | 8.0 | .80 | " | |

Mode 3: Variable Geometry Secondary and Fixed Geometry Primary Air

On completion of Mode 2 testing, efforts were directed toward optimizing performance. The primary goal at this point was to equal or exceed the power output of the normal spark ignition engine. Theoretically[7] and experimentally[8], this goal could be accomplished with lower specific fuel consumption and decreased wasteful exhaust emissions.

Analysis of Mode 2 performance (see section VII) showed a trend in which the heat balanced engine approached the power output of the standard engine. To further investigate this trend, a pressure exchange cap was designed from the best Mode 1 and 2 results, 0.080 inch edge gap clearance, a $\beta$ of 0.80 and a compression ratio of 8.0:1. Preliminary testing indicated that both power output and specific fuel consumption were slightly improved over peak Mode 2 results. However, output levels of the standard engine were not realized.

Further analysis of optimization test data indicated that maximum power was obtained at an air/fuel ratio slightly richer than stoichiometric. Previous results[1] indicated best power occurred at about stoichiometric composition. This indicated the need for more secondary air. The fixed secondary air orifice (copper diaphragm) was removed from the union coupling. Airflow through the secondary air intake manifold was now controlled by a gate valve located at the upper end of the secondary air inlet pipe (fig. VI-4). Adjustment of this valve allowed control of the secondary air entering the engine. This modification was the major change defining Mode 3.

Many test variations combining secondary air, primary air choke, and fuel level were tried with the result that power output was demonstrated to be directly related to the engine's ability to ingest the richest possible

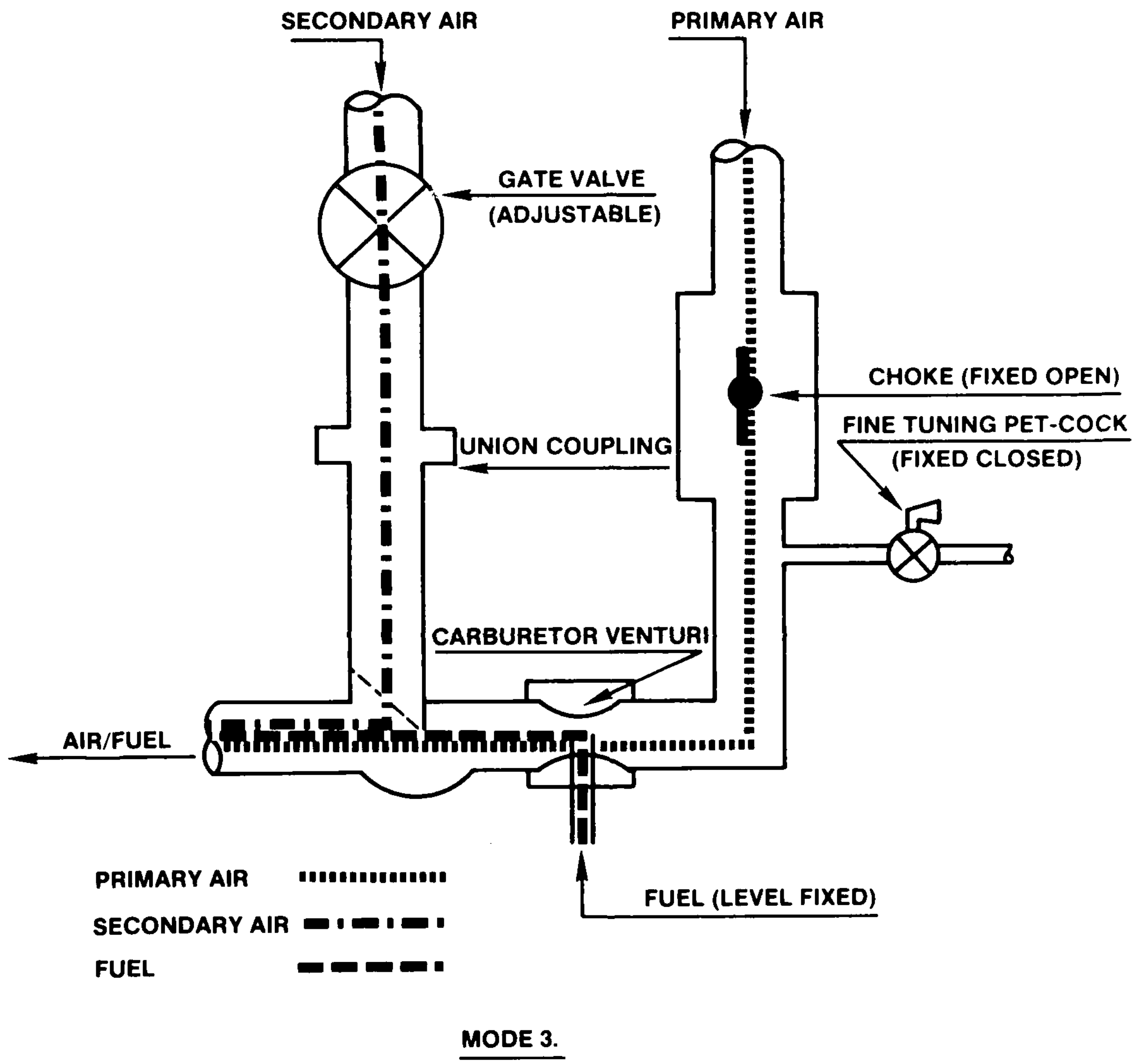



**Figure VI-4**
**Schematic of Primary and Secondary Air Control**

primary charge of air-fuel late in the intake stroke and to receive an early, extremely lean mixture of secondary air. Using this method maximum power was obtained at a overall air/fuel ratio slightly leaner than stoichiometric. In effect, the limiting case of ingesting a rich fuel-air charge into a combustion chamber filled with only air is the concept of fuel injection. This area of research will be examined at a later date.

Mode 3 operating procedures were established using fixed primary and variable secondary air. The engine was run at open throttle and the choke secured in a wide open position. The adjustable carburetor bowl assembly was raised to its upper limit until the fuel was level with the venturi centerline, a fuel height of 0.0 inches. This operating mode made it possible to obtain a late, rich primary charge having sufficient velocity to enter the combustion chamber. Earlier attempts at using the choke to enrich the mixture were less successful due to low primary air velocities. Secondary air was adjusted until peak engine performance was obtained. It should be noted that the amount of secondary air was the only variable in Mode 3 operations.

Test results using these procedures indicated that: engine maximum output was approximately 38% over the standard engine maximum taken at equal RPM and compression ratio; specific fuel consumption was reduced by about 10% at best economy and about 30% at best power; wasteful exhaust gas emissions were significantly reduced.

## VII. COMPARISON OF RESULTS

### OTTO and Mode 1 Behavior

The standard and modified engine were run over 3 (three) RPM, three fuel grades, and three compression ratios. These results and all data are fully documented[9] for all modes discussed in this report.

Analysis of results in Figures VII-1 through 7 shows that the expectations of Reference 1 were not attained in this mode. Figure VII-4 shows the general trend expected but output is low and consumption is high (The curves presented are typical of Mode 1). One would be tempted to say that the engine was behaving as a "lean" Otto after examination of the air-fuel ratio. Standard techniques[10] for reduction of dynamometer data were used.

Because of the extension of the scope of experimenting to other air/fuel modes, testing with other than regular grade fuel was dropped at this point and the parameters for Mode 2 were established. Further reports will cover the multifuel capability of the engine.

### Mode 2 Behavior

To "capture" the heat balanced cycle one must have not only good insight into the function of the two chamber interaction in the cylinder but also the function of the air-fuel system. For the theory[6,8] to work in practice, good separation of charge must occur, in fact, we like to refer to the engine as a separated charge rather than a stratified charge engine. As described earlier the orifice used in the secondary air manifold was moved from the extreme discharge end of the manifold to an intermediate position and decreased in size since the region between the discharge end and the orifice

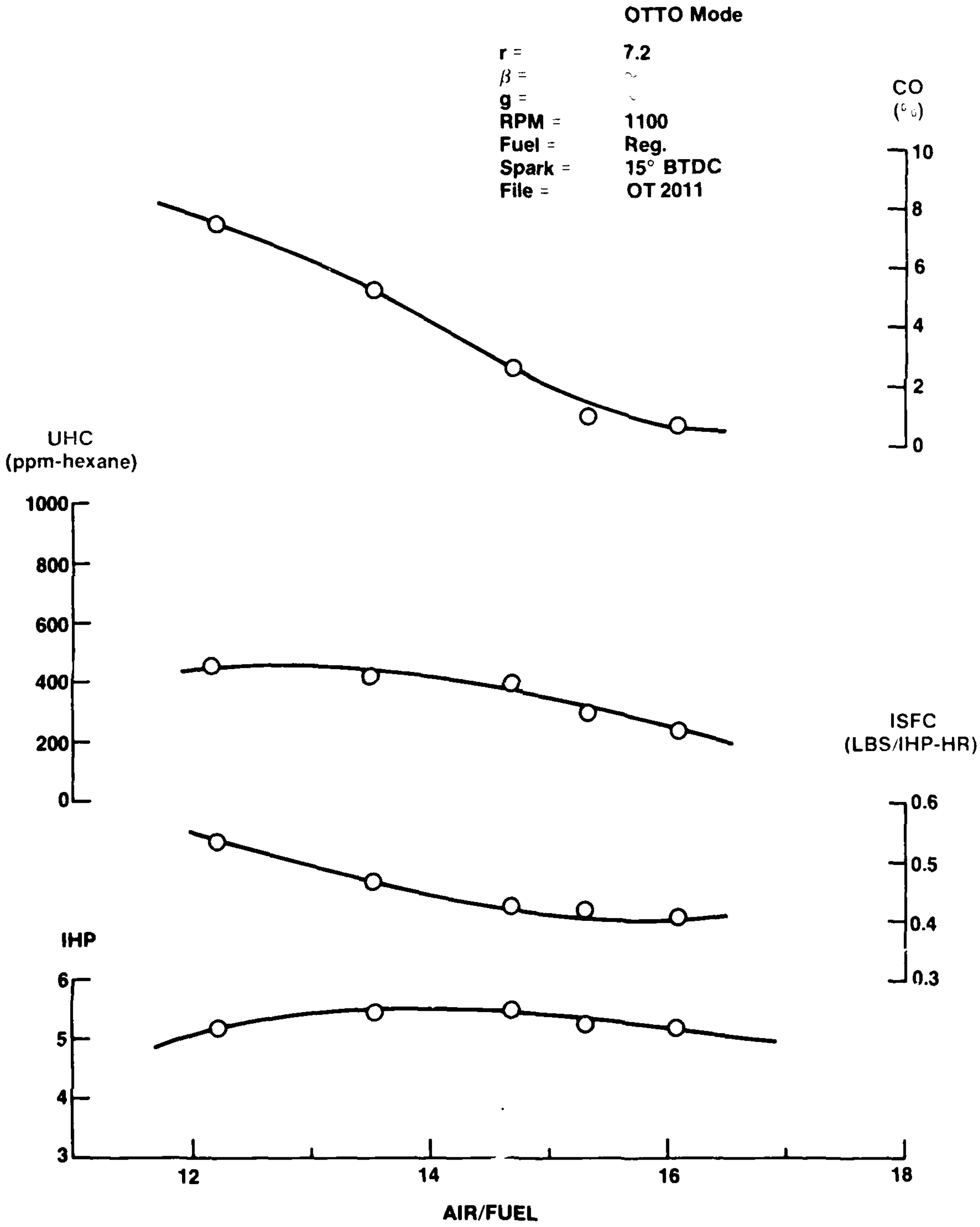



**Fig. VII-1**
**OTTO Engine Performance**

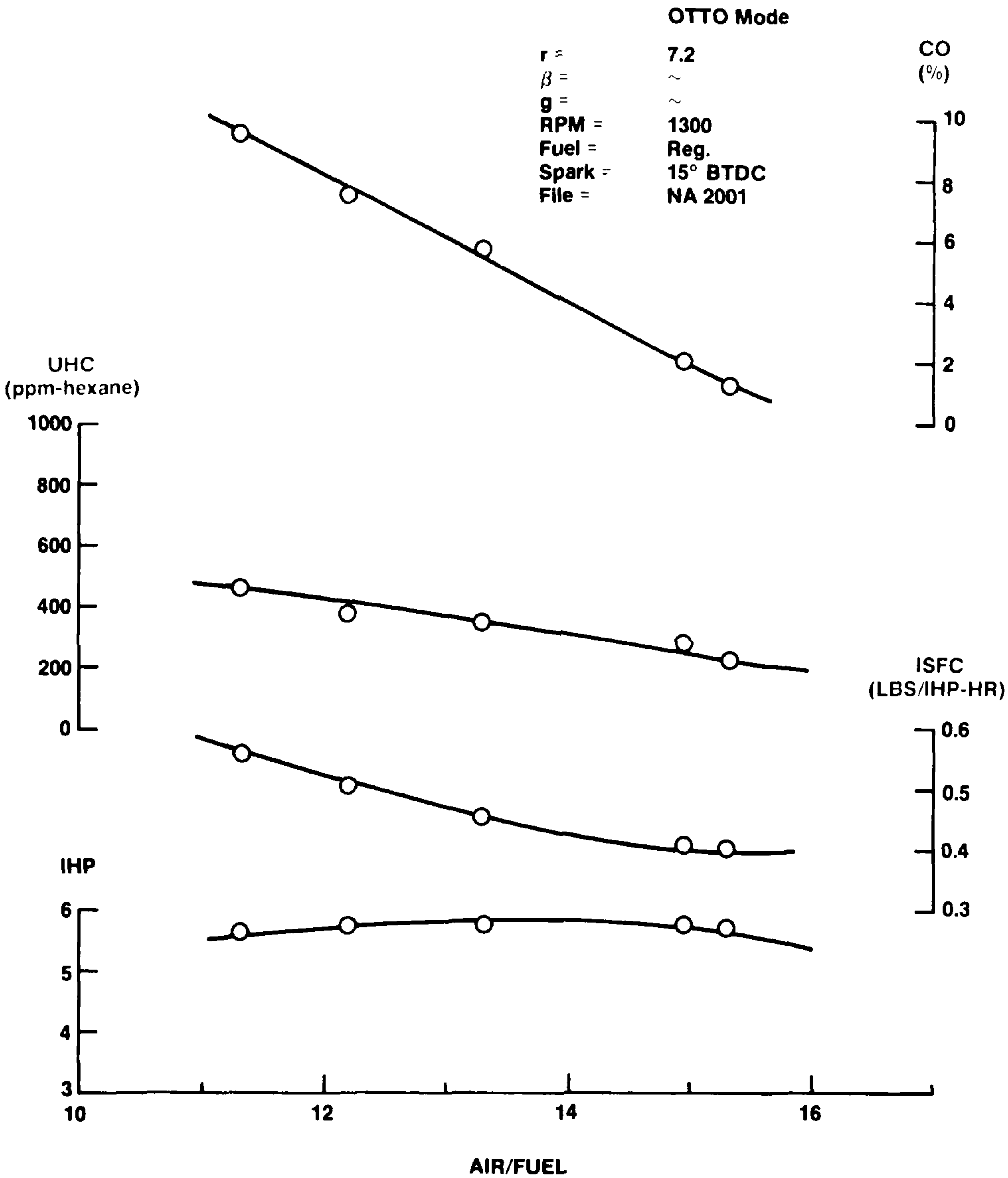



**Fig. VII-2**
**OTTO Engine Performance**

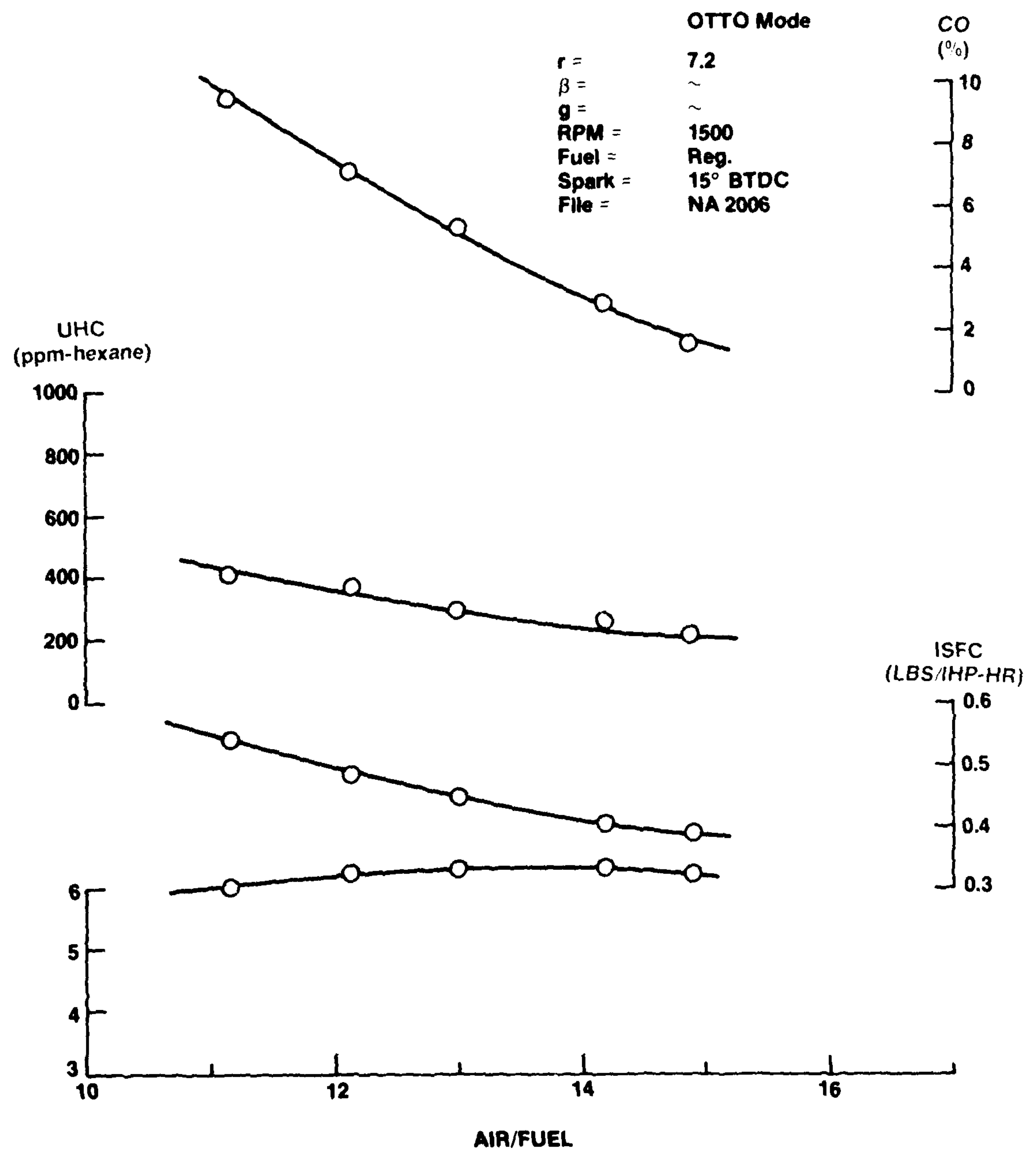



**Fig. VII-3**
**OTTO Engine Performance**

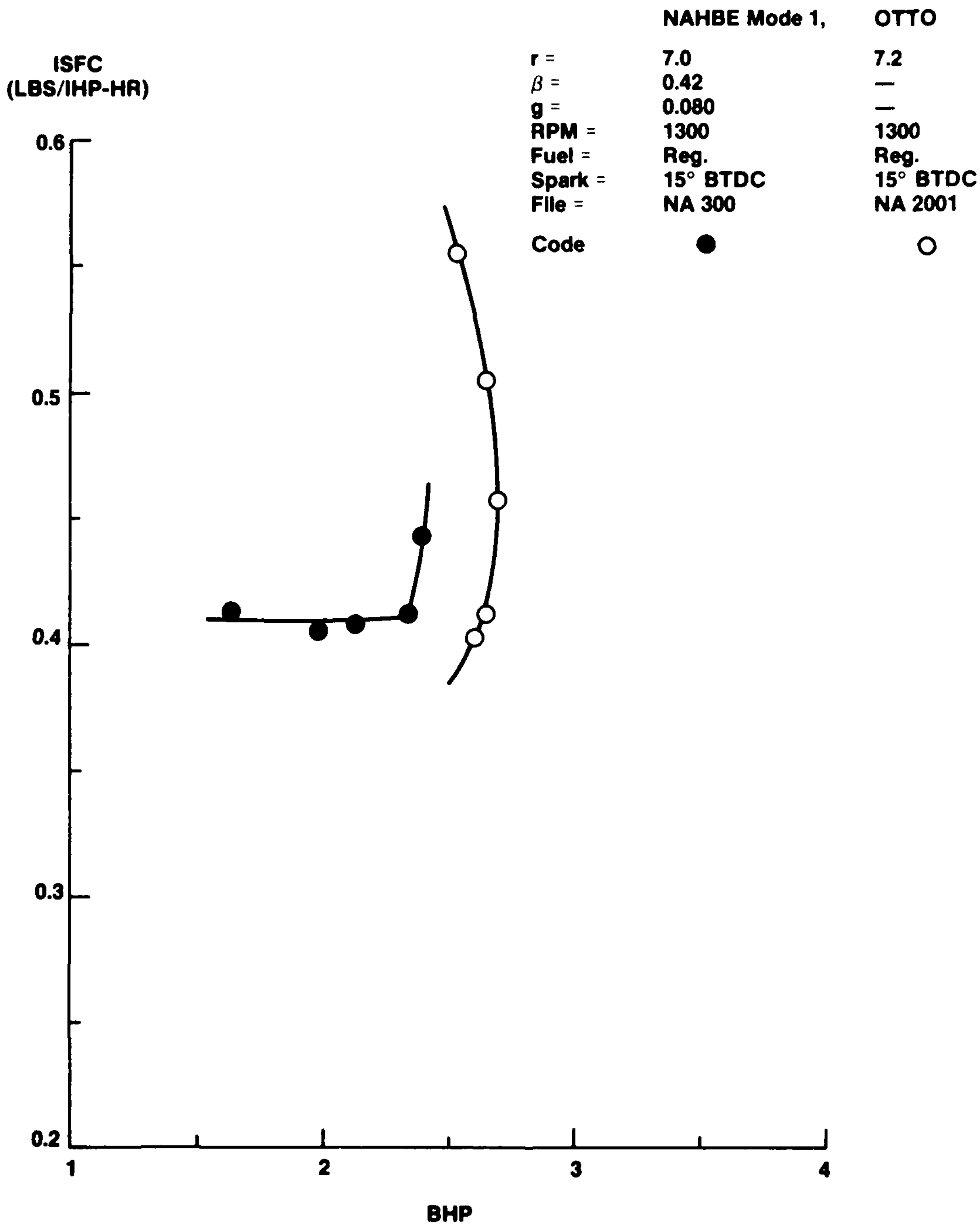



**Fig. VII-4**
**OTTO, NAHBE — Mode 1 Comparison**

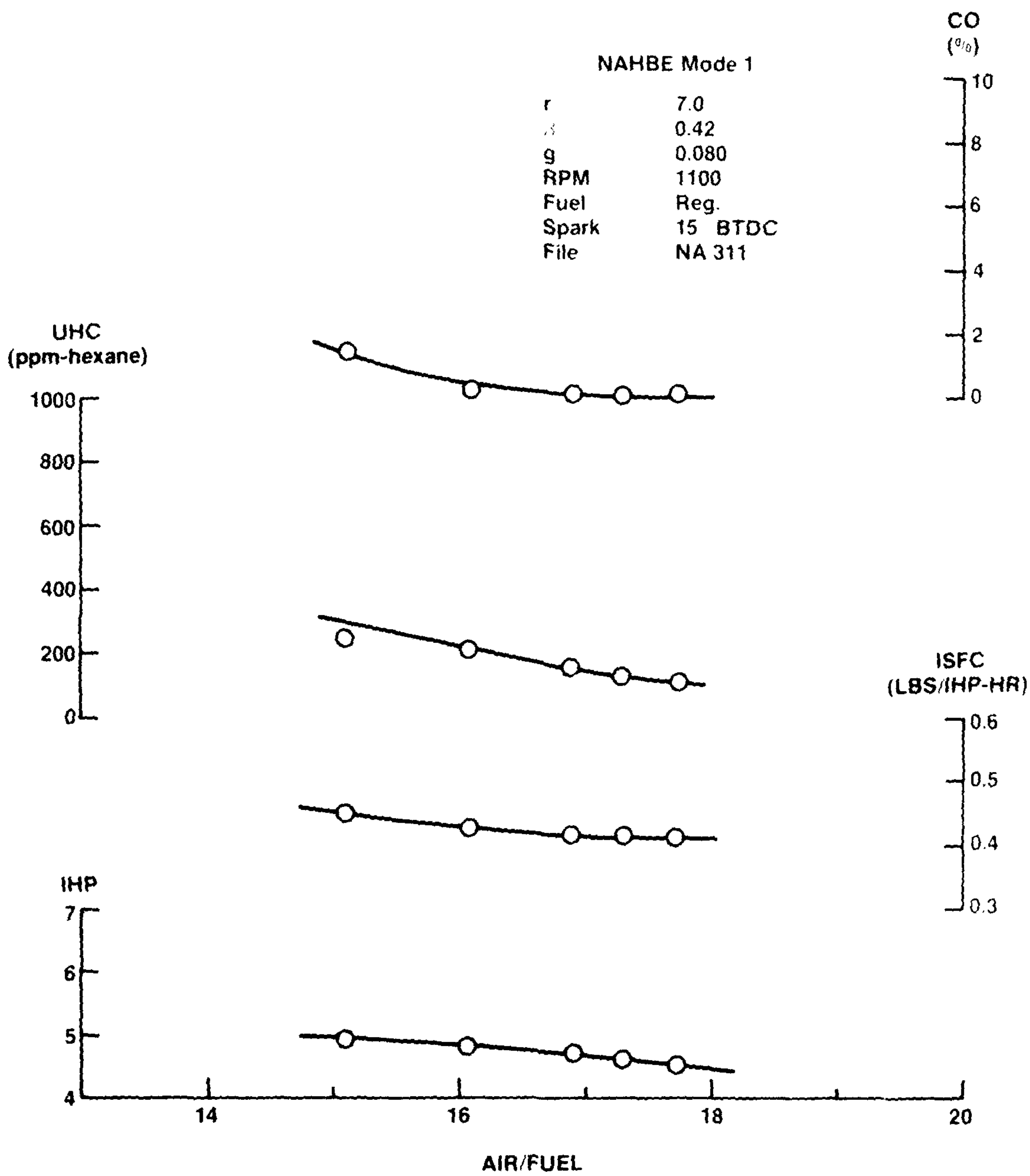



**Fig. VII-5**
**NAHBE Mode 1 Performance**

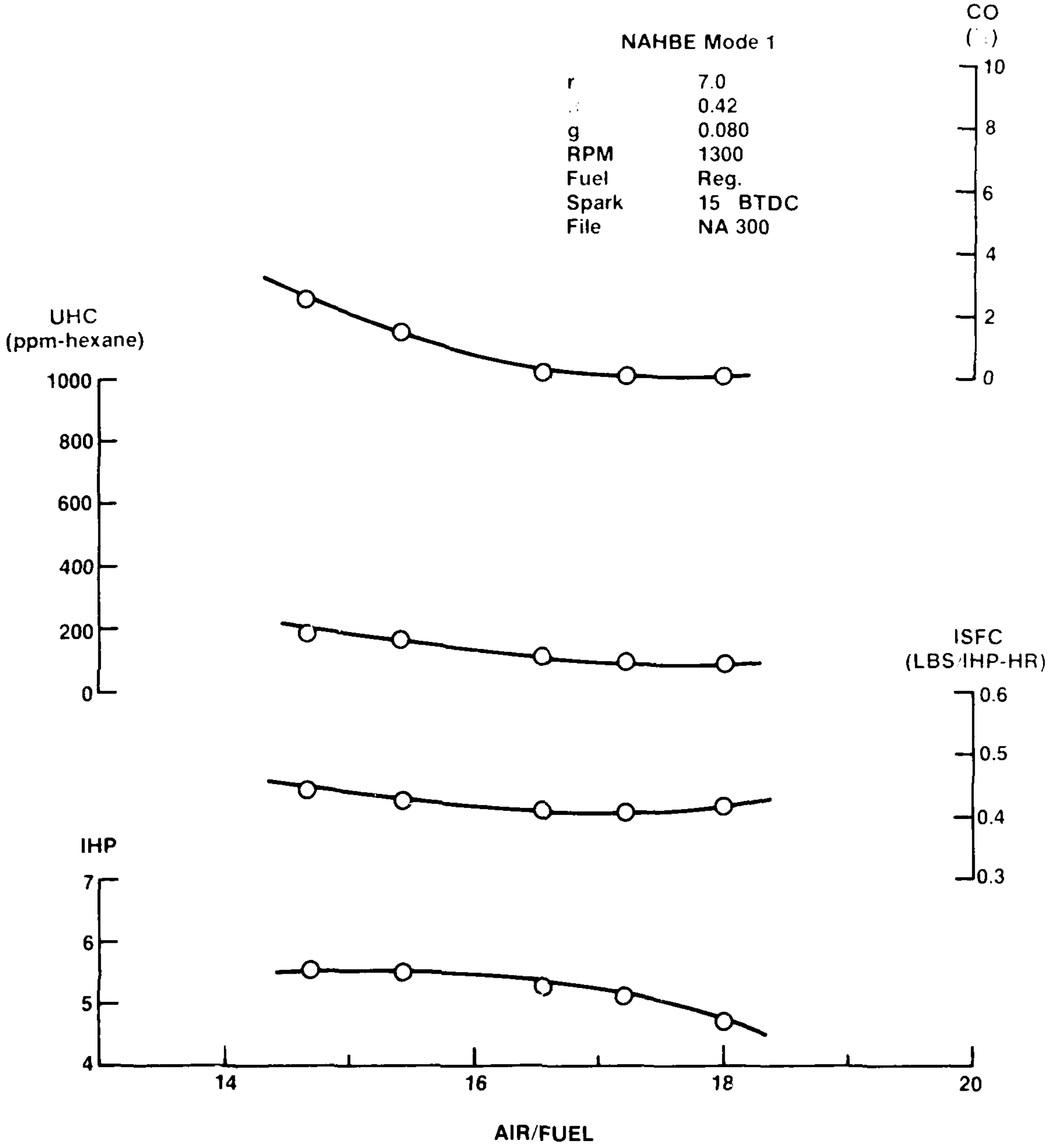



**Fig. VII-6**
**NAHBE Mode 1 Performance**

## Table VII-1

## OTTO PARAMETRIC EVALUATION
## SUMMARY OF RUN CONDITIONS

| FUEL TYPE | 3 (LOW TEST) | | | 2 (MID RANGE) | | | 1 (HIGHEST) | | |
|---|---|---|---|---|---|---|---|---|---|
| r | 5.6 | " | " | " | " | " | " | " | " |
| β | — | | | | | | | | |
| GAP | — | | | | | | | | |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | OT090 | OT080 | OT085 | OT060 | OT050 | OT055 | OT075 | OT065 | OT070 |
| r | 7.2 | | | | | | | | |
| β | — | | | | | | | | |
| GAP | — | | | | | | | | |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | OT041 | OT031 | OT036 | OT2011 | NA2001 | NA2006 | OT026 | OT2016 | OT021 |
| r | 8.0 | | | | | | | | |
| β | — | | | | | | | | |
| GAP | — | | | | | | | | |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | OT140 | OT130 | OT135 | OT110 | OT100 | OT105 | OT125 | OT115 | OT120 |

## Table VII-2

## NAHBE PARAMETRIC EVALUATION, MODE 1
## SUMMARY OF RUN CONDITIONS

| FUEL TYPE | 3 (LOW TEST) | | | 2 (MID RANGE) | | | 1 (HIGHEST) | | |
|---|---|---|---|---|---|---|---|---|---|
| r | 5.5 | " | " | " | " | " | " | " | " |
| β | .30 | " | " | " | " | " | " | " | " |
| GAP | .080 | " | " | " | " | " | " | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA256 | NA242 | NA249 | NA214 | NA200 | NA207 | NA235 | NA221 | NA228 |
| r | 7.0 | " | " | " | " | " | " | " | " |
| β | .42 | " | " | " | " | " | " | " | " |
| GAP | .080 | " | " | " | " | " | " | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA341 | NA331 | NA336 | NA311 | NA300 | NA306 | NA326 | NA316 | NA321 |
| r | 7.8 | " | " | " | " | " | " | " | " |
| β | .50 | " | " | " | " | " | " | | " |
| GAP | .080 | " | " | " | " | " | " | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA040 | NA030 | NA035 | NA110A | NA100A | NA105A | NA125 | NA115A | NA120 |

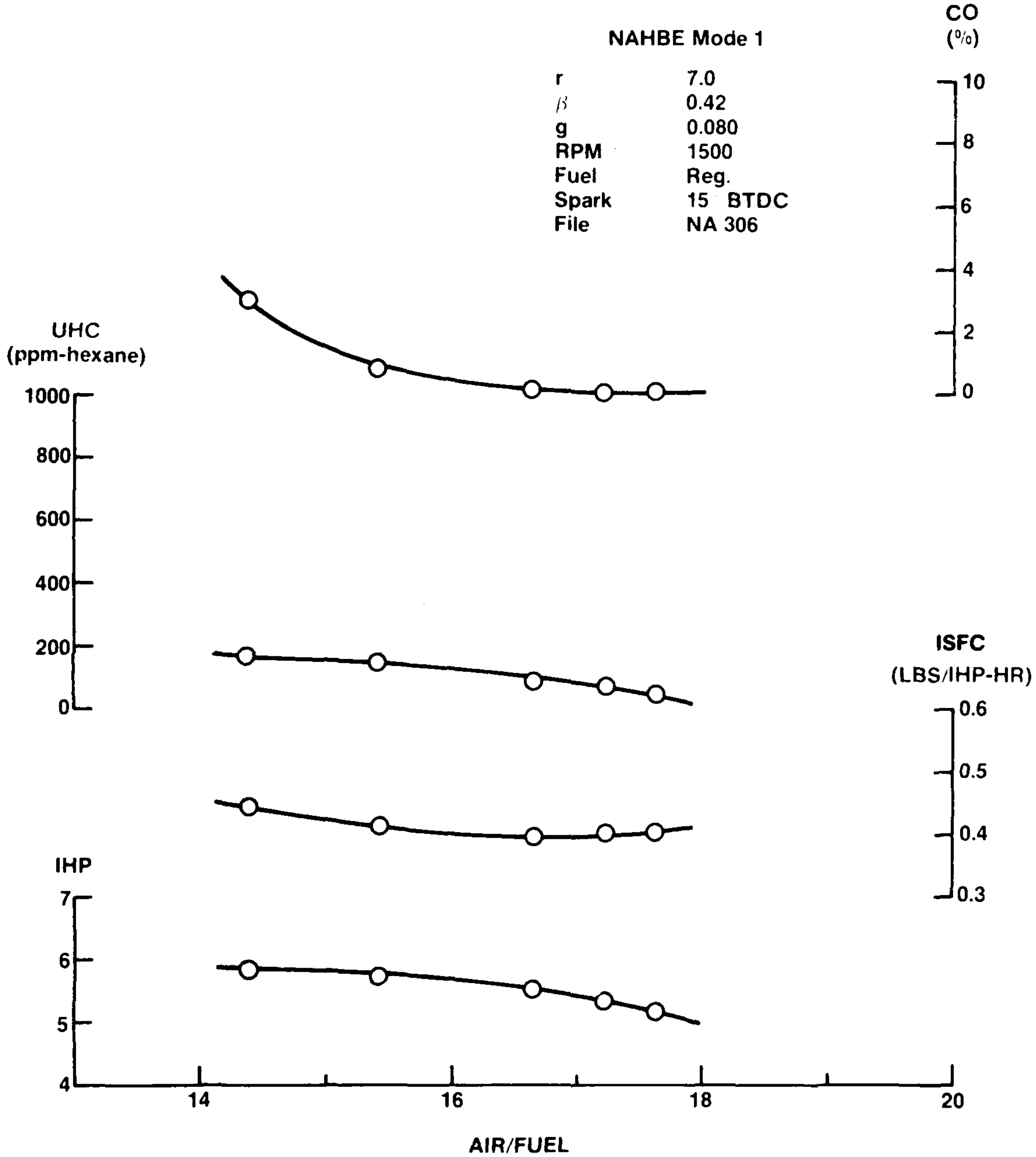



Fig. VII-7
NAHBE Mode 1 Performance

actes as an accumulator chamber (see Figure VI-3). The engine manifold pressure always being lower than atmospheric, the accumulator continuously charges with air and discharges only on opening of the intake valve. This proved to be semi-effective in creating a separated charge while Mode 1 was less effective.

It should become evident in the course of this comparison that Mode 1 behavior is one extreme of heat balanced operation, (the poorest), Mode 3 the opposite extreme (best) while Mode 2 is an intermediate condition, though there are some good features to both Mode 1 and 2.

Typical results of this mode of operation are seen in Figures VII-8 through 15. Output at 1300 RPM exceeded the Otto for the same compression ratio while output at 1500 RPM did not. The characteristic "U" shaped curve of the heat balanced cycle[1], figure VII-9 was approached, however. The right vertical corresponds to rich fuel composition; the left vertical to lean. Stoichoimetric composition (overall, that is primary plus secondary air) is generally just above the right corner of the "U" (point *), so the normal operating mode is at essentially constant specific fuel consumption (lower leg of the "U").

It can not be determined from calculated results alone what caused increasing richness (generally) of the mixture after a certain fuel level was reached. Recall, however, that the runs began with a fuel level two inches below the venturi throat and the level was descreased to five inches. At the same time, the inlet throttle plate ("choke") was closed slightly, tuning the inlet to allow for decreased fuel height. The fluctuating inlet manifold pressure was not recorded though it must have decreased with closure of the inlet throttle plate. Pressure drop across the atmospheric

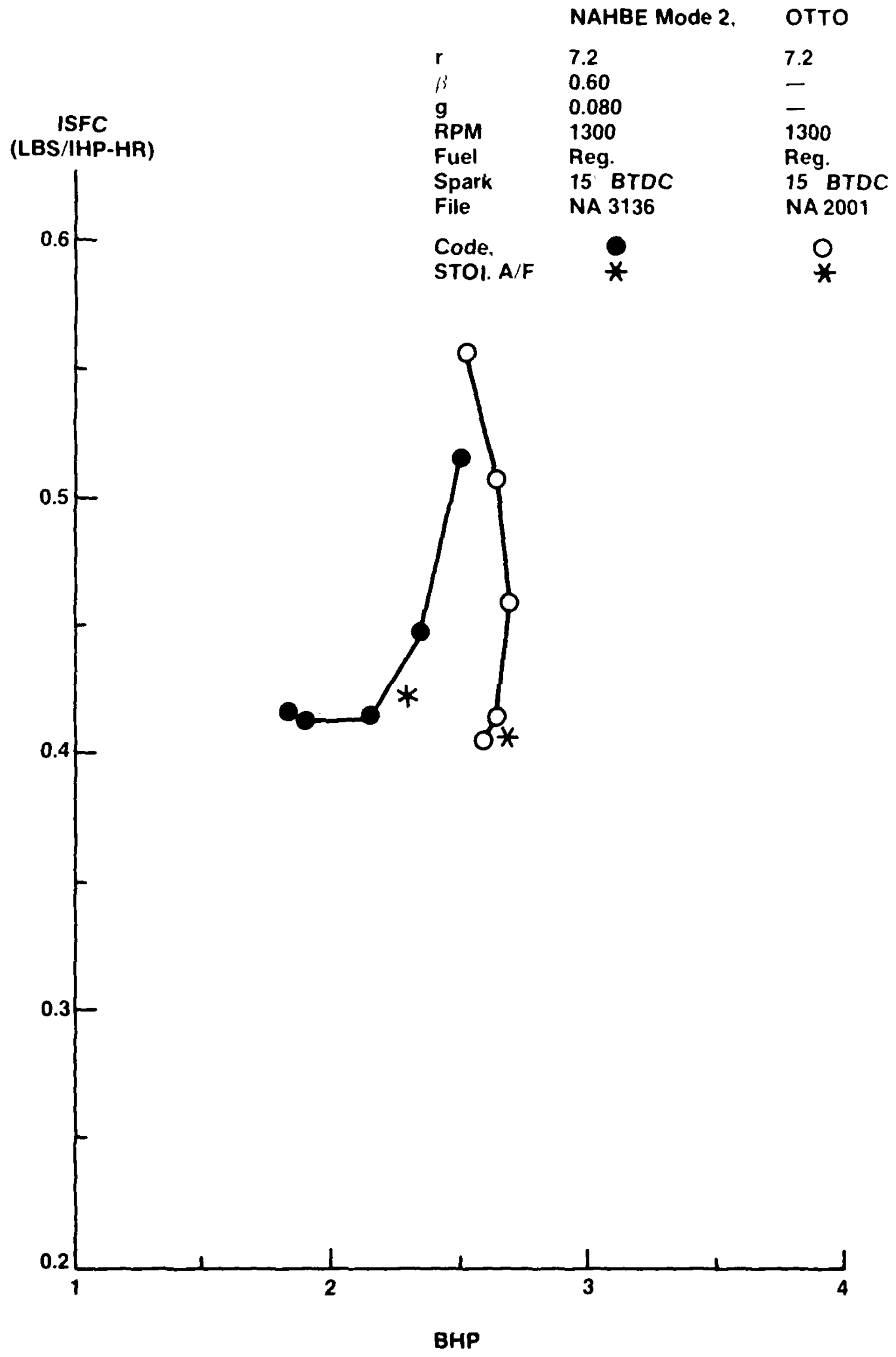



Fig. VII-8
OTTO, NAHBE — Mode 2 Comparison

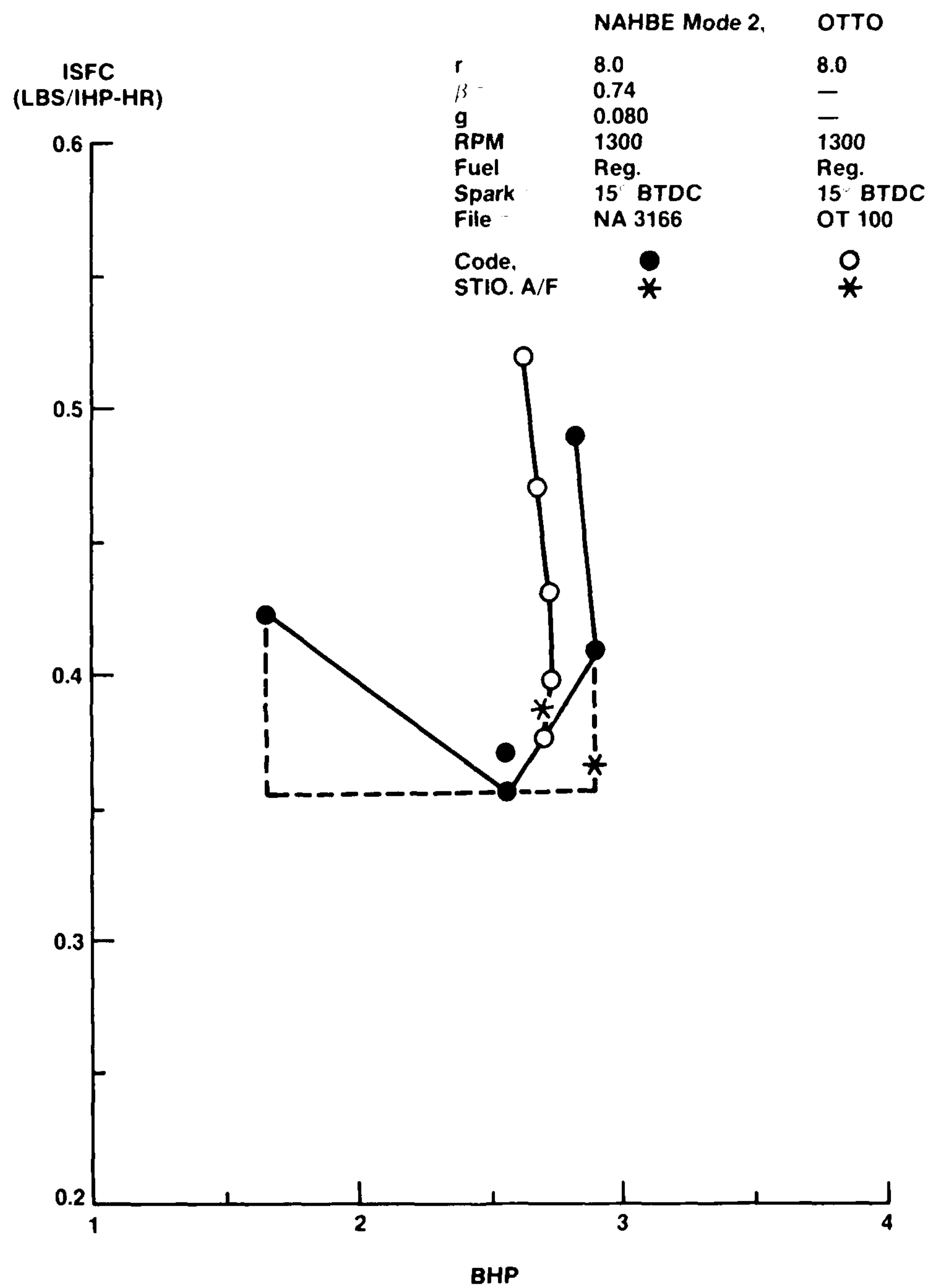



Fig. VII-9
OTTO, NAHBE — Mode 2 Comparison

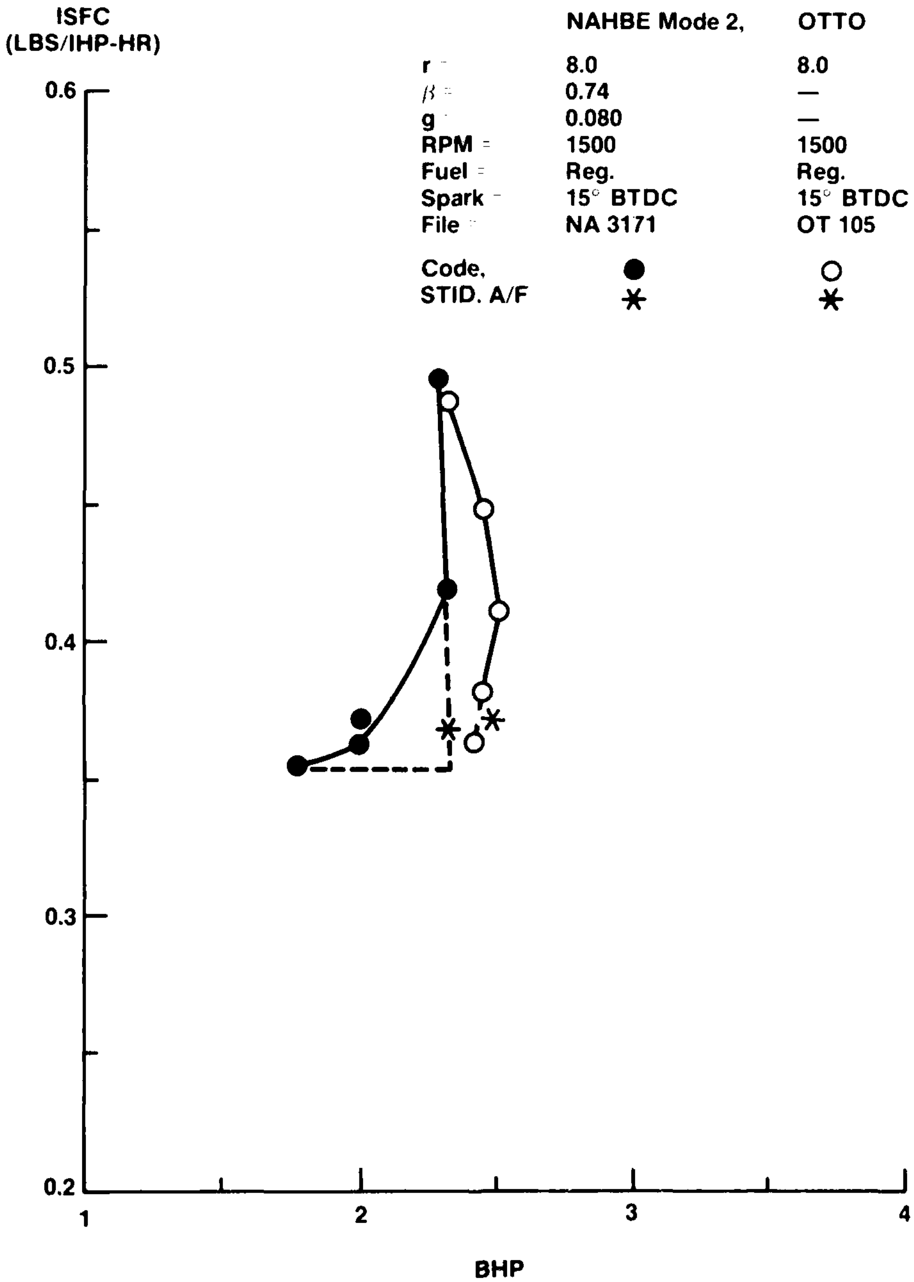



Fig. VII-10
OTTO, NAHBE — Mode 2 Comparison

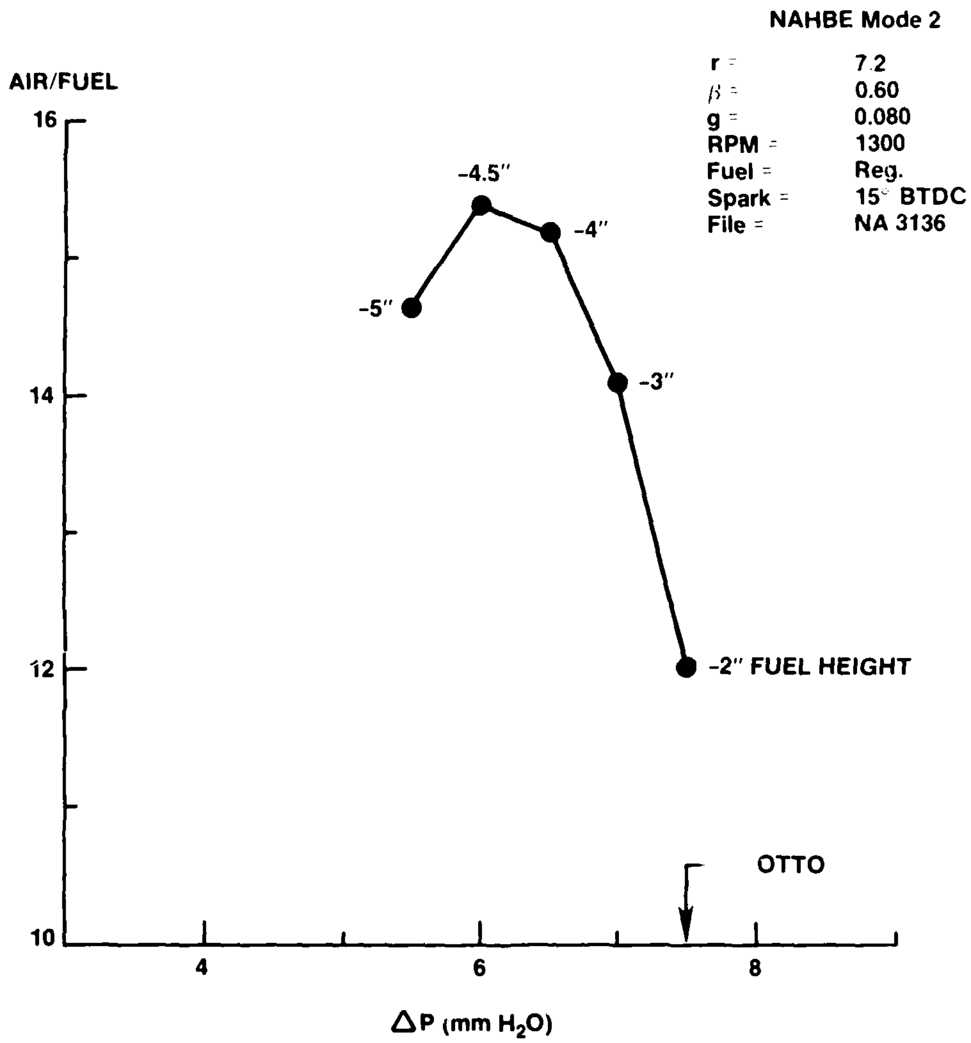



**Fig. VII-11**
**Air Supply Metering Orifice Pressure Drop**

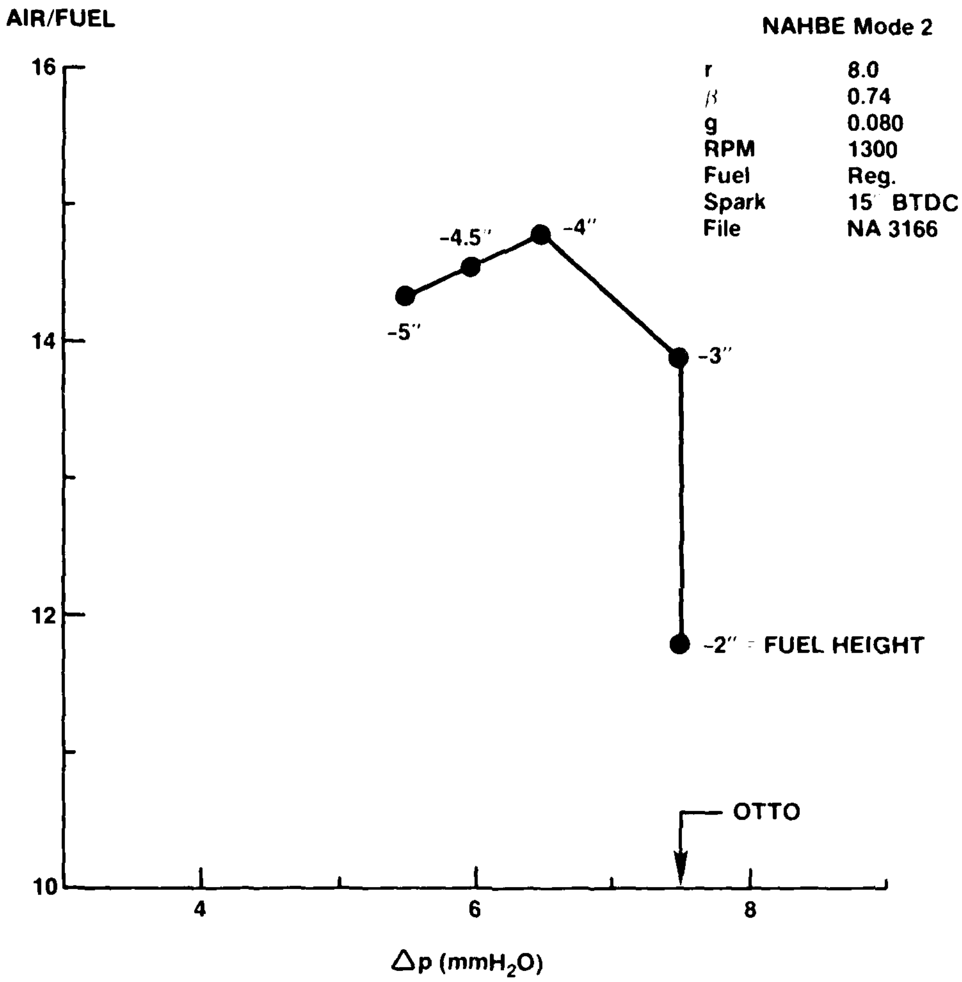



**Fig. VII-12**
**Air Supply Metering Orifice Pressure Drop**

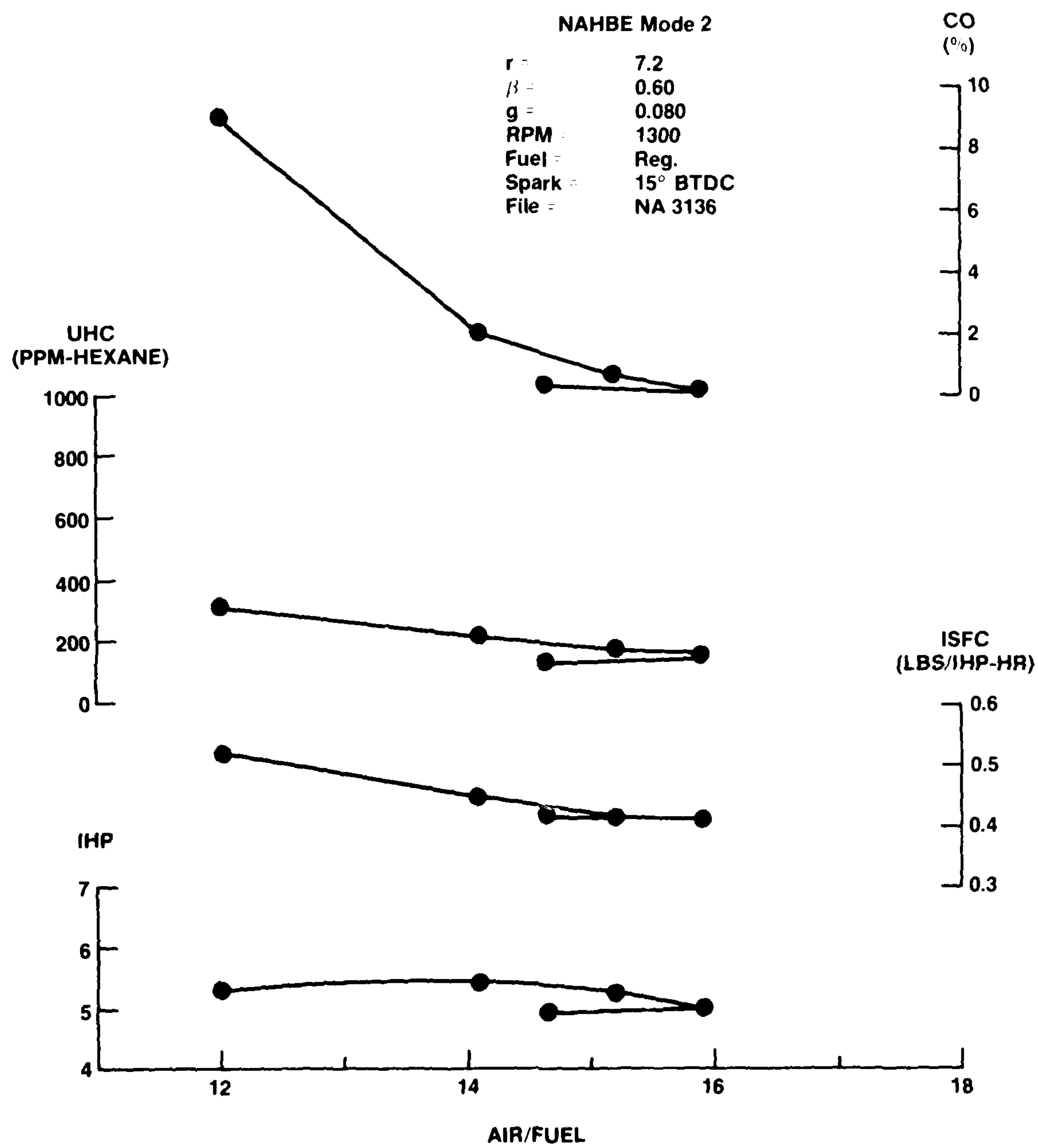



Fig. VII-13
NAHBE — Mode 2 Performance

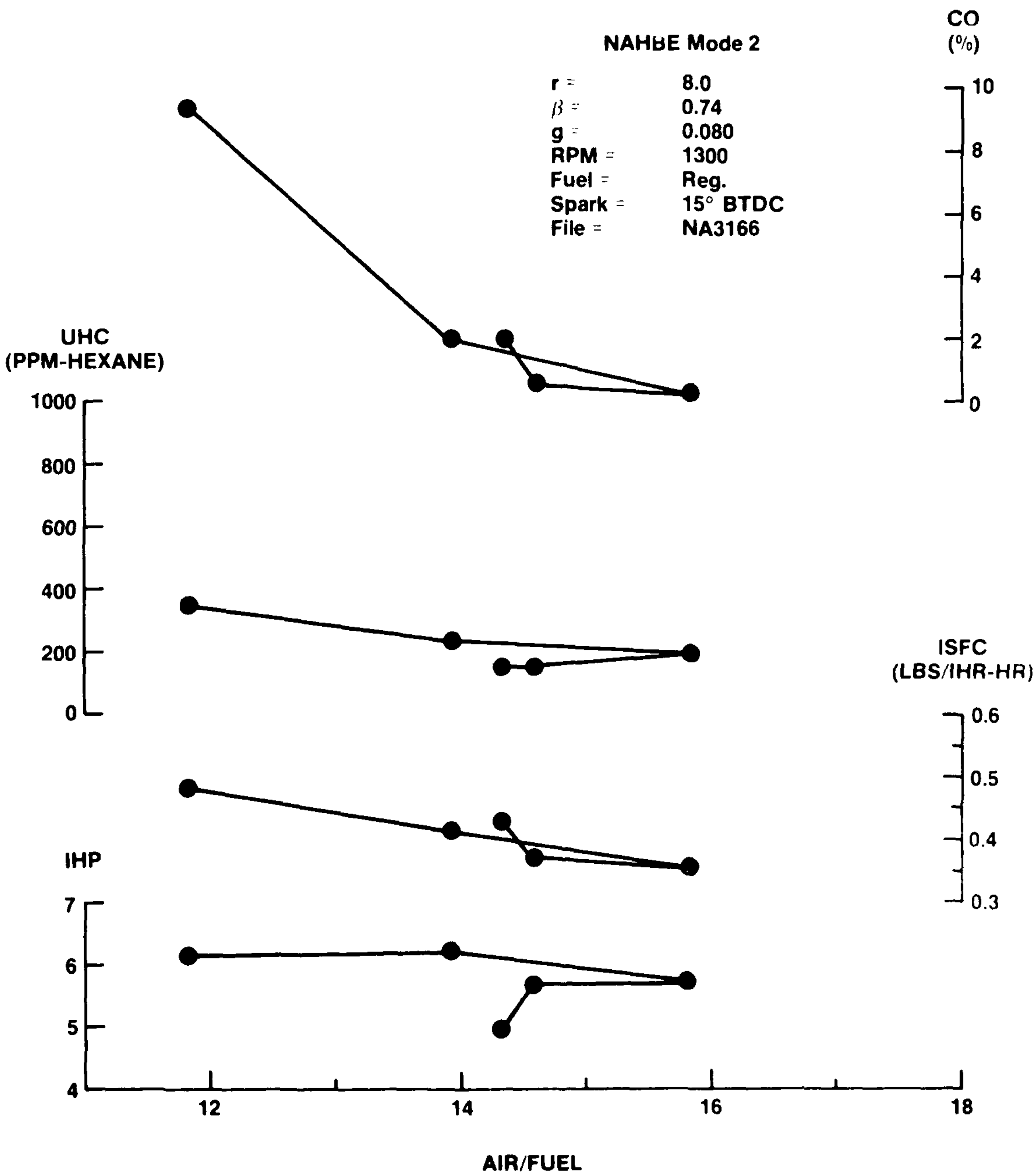



**Fig. VII-14**
**NAHBE — Mode 2 Performance**

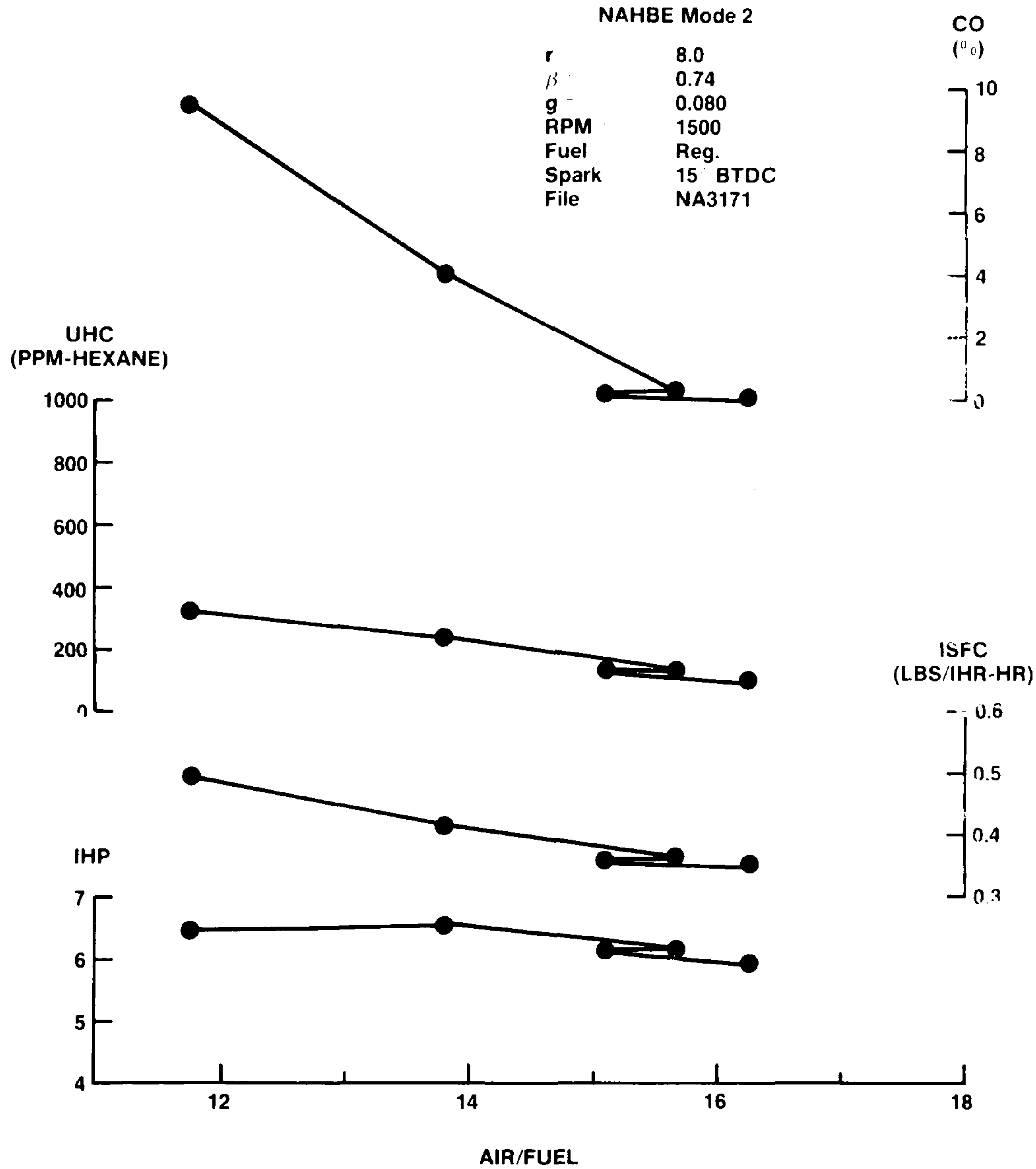



Fig. VII-15
NAHBE — Mode 2 Performance

air supply was recorded and it showed a significant decrease on closure of the inlet throttle plate (see Fig, VII-10). Air flow was therefore decreased as fuel height was decreased, with the air-fuel ratio reaching a maximum then decreasing. This method of operation was avoided in Mode 3.

## Variation of Clearance Gap

Attention is turned now to variation of the gap width between the cap and cylinder wall. The run matrix for this mode - Table VII-3 shows the total number of combinations tested.

Typical results are given in Figs. VII-16 and 17. Evaluation of calculated results meant comparison of Heat Balanced output with OTTO output, emissions, and fuel consumption for each gap width for three compression ratios. To reduce the labor of evaluation of these results the maximum indicated horsepower was divided by the indicated specific fuel consumption, the unburned hydrocarbons and percent carbon monoxide in what is proposed as the Run Quality Index (RQI),

$$RQI = \frac{IHP \cdot (10^3)}{ISFC \cdot UHC \cdot CO} \quad \text{-----} \quad \text{VII-1}$$

where specific fuel consumption is in lbs/hphr, unburned hydrocarbon (hexane) in parts per million (ppm) and CO in percent. This compresses results of say fig. VII-13 into a single curve. Since reduction in labor was sought no attempt to "clean up" the equation - dimensionally - was made. The factor $10^3$ was inserted to give a reasonable range of values. One could interpret this as an overall or gross efficiency since it amounts to output divided by "net" input.

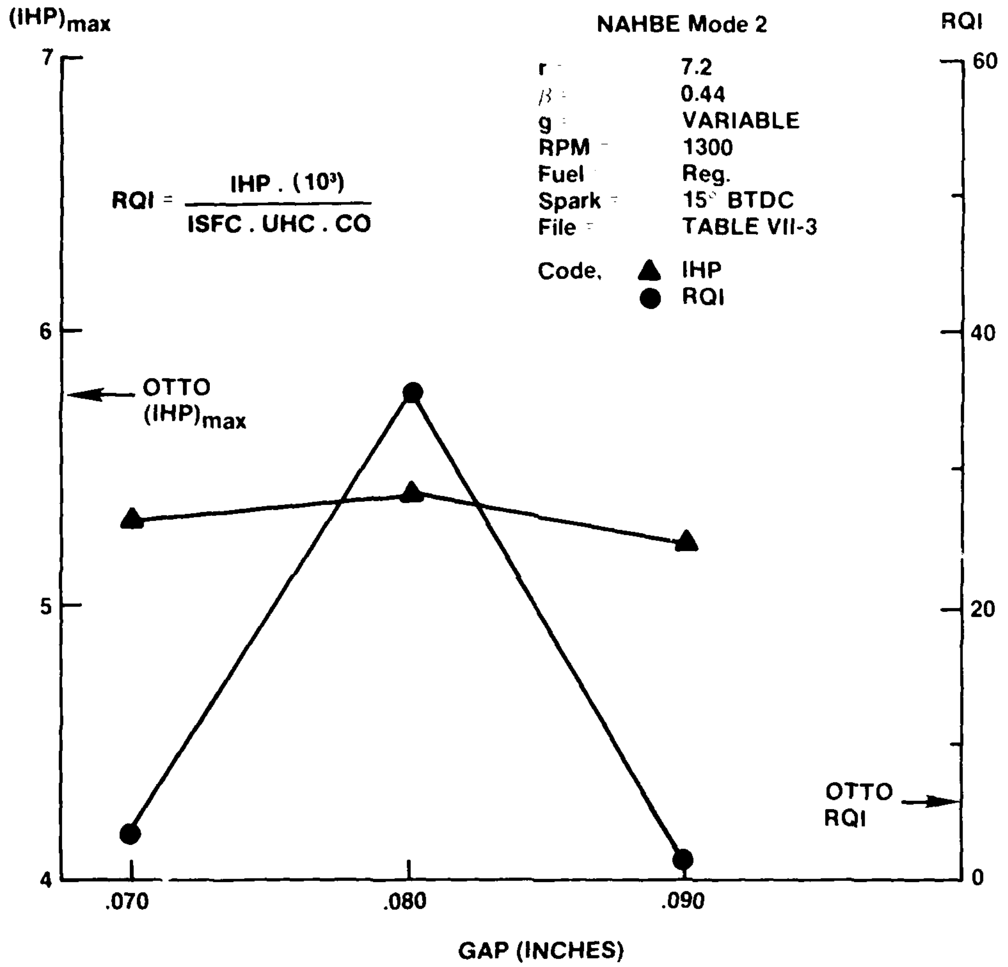



Fig. VII-16
Performance Variations with Varying Gap Width

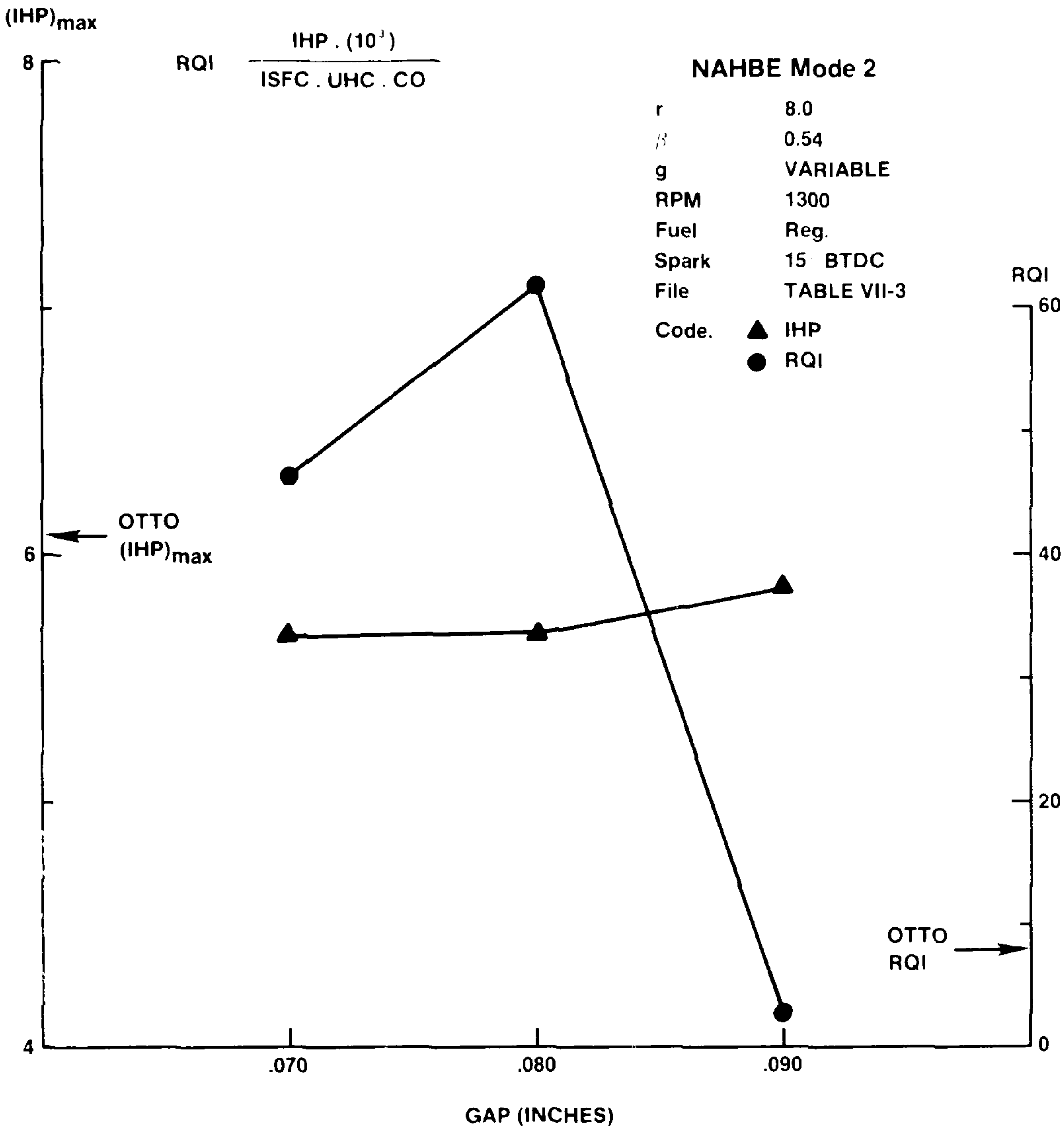



Fig. VII-17
Performance Variations with Varying Gap Width

Not all plots were as marked in RQI at an 0.080 gap as those shown but the trend was evident throughout; also, full heat balanced operation would not be demonstrated until Mode 3. From this point on the 0.080 gap was the only one used. More will be said about the RQI in Mode 3 operation.

Variation of Balancing Chamber Volume

Two parallel developments converged in this experimental phase. While the experiments were in progress one of the predictions of the air standard Heat Balanced Cycle analysis, namely, there is an optimum balancing ratio, was examined for fuel-air mixtures. The maximum in performance was found to occur again at a $\beta$ of about 0.8 for overall stoichiometric composition at a compression ratio of 8:1 at 1300 and 1500 RPM.

Analysis of the experimental results in Figs. VII-18-20 for these condition shows a similar trend. This may be pure coincidence since real engine operation is non-steady and non-equilibrium in nature while the idealized theremodynamic analysis is for quasi-equilibrium model. Further testing was with balancing ratios in the neighborhood of 0.80.

Peak Pressure Behavior

Piezo electric pressure transducer recordings for each run were recorded on a storage type scope and peak pressure scaled off. Plots at three compression ratios for various values of $\beta$ are given in Figures VII-21 23. There is significant pressure reduction from that of the OTTO for all balancing ratios except those at $\beta$ values near 1.0 at the two higher compression ratios. It appears that peak pressure is related to $\beta$ for fixed secondary air but the relationship is non-linear and uncertain due to manifold pressure changes steming from the primary air choke. In Mode 3

with variable secondary air, higher peak pressures than the OTTO were observed with a corresponding increase in output. Sample scope traces for the data of Fig. VII-24 are given in Fig. VII-33.

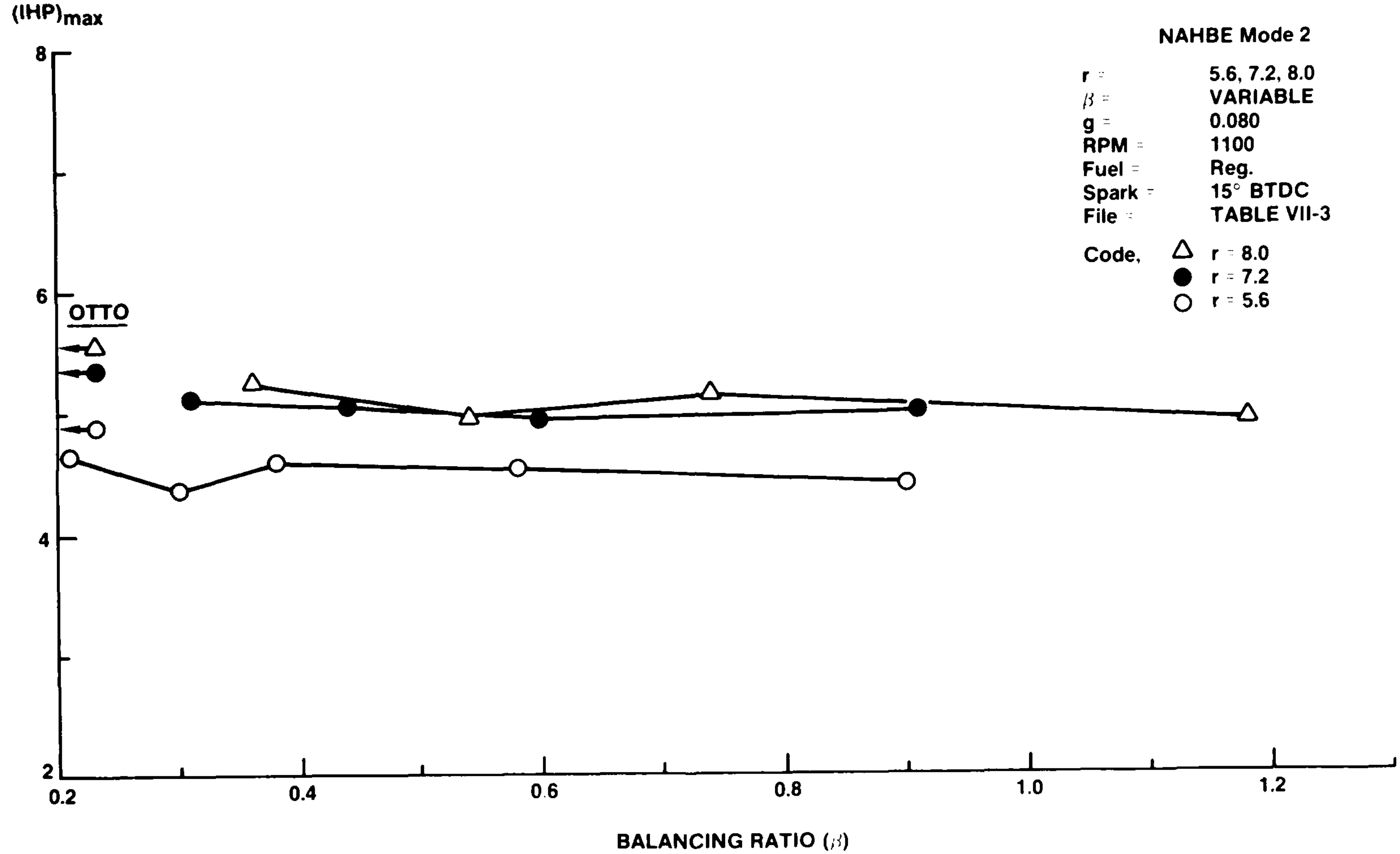



Fig. VII-18
Variation in Maximal Output with Balancing Ratio

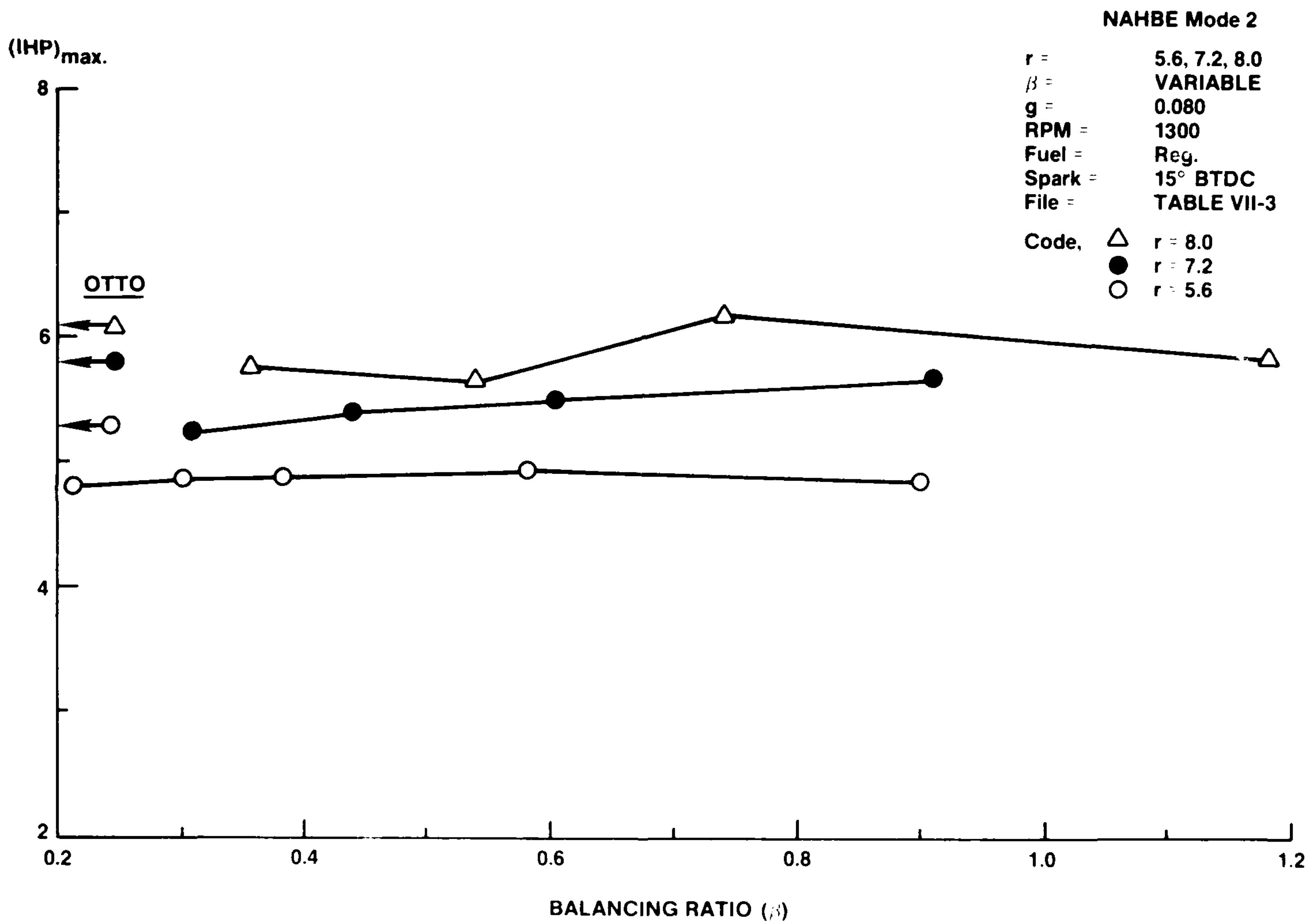



Fig. VII-19
Variation in Maximum Output with Balancing Ratio

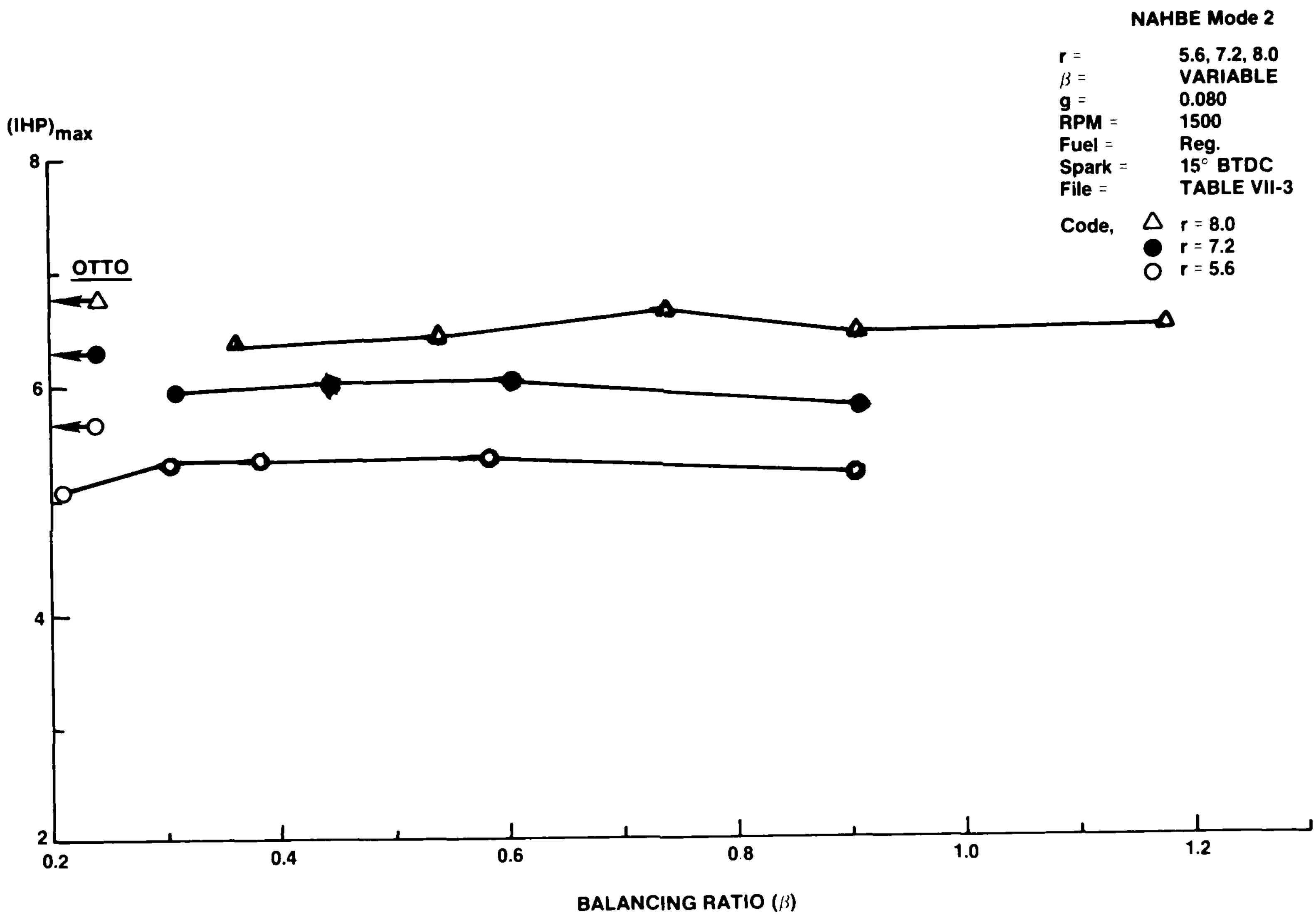



Fig. VII-20
Variation in Maximum Output with Balancing Ratio

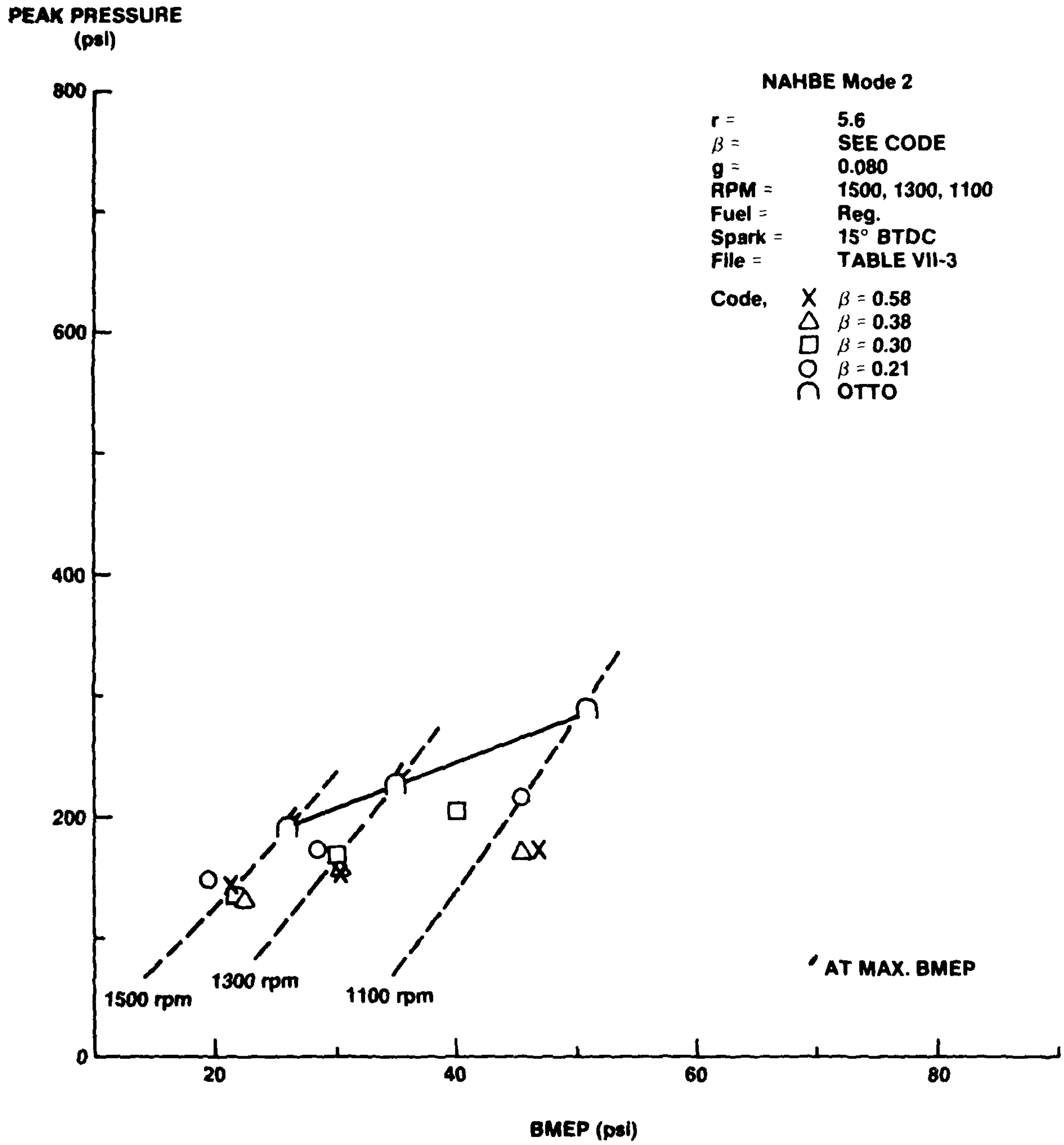



**Fig. VII-21**
**Peak Pressure Variation with Balancing Ratio**

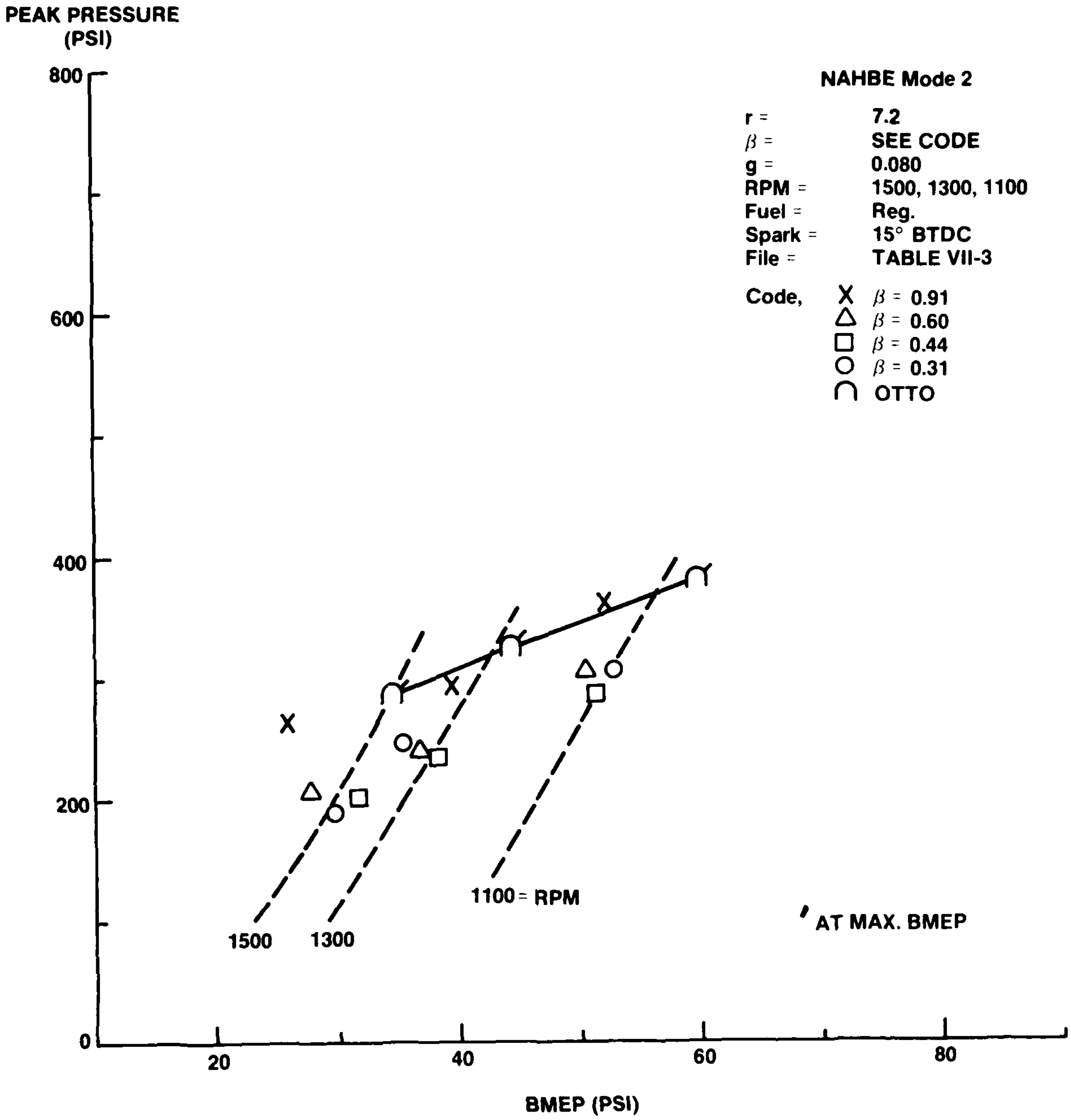



Fig. VII-22
Peak Pressure Variation with Balancing Ratio

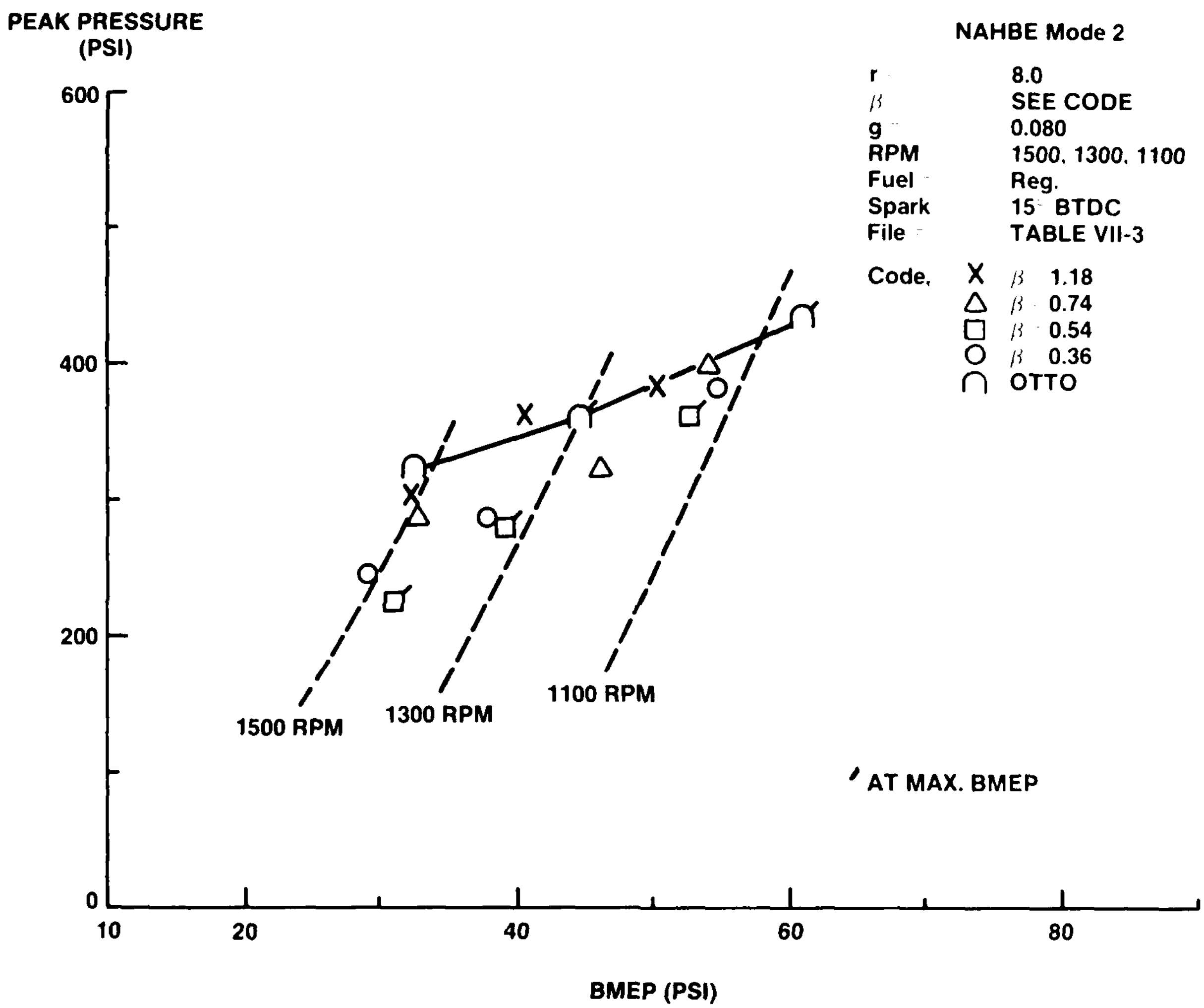



Fig. VII-23
Peak Pressure Variation with Balancing Ratio

# Table VII-3

## NAHBE PARAMETRIC EVALUATION, MODE 2
## SUMMARY OF RUN CONDITIONS

| | FUEL TYPE 3 (LOW TEST) | | | 2 (MID RANGE) | | | 1 (HIGHEST) | | |
|---|---|---|---|---|---|---|---|---|---|
| r | | | | 5.6 | " | " | | | |
| β | | | | .21 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3251 | NA3241 | 3246 | | | |
| r | 5.6 | " | " | " | " | " | " | " | " |
| β | .30 | " | " | " | " | " | " | " | " |
| GAP | .070 | " | " | .080 | " | " | .090 | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA3071 | NA3061 | NA3066 | NA3026 | NA3016 | NA3021 | NA3116 | NA3106 | NA3111 |
| r | | | | 5.6 | " | " | | | |
| β | | | | .38 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3161 | NA3151 | NA3156 | | | |
| r | | | | 5.6 | " | " | | | |
| β | | | | .58 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3206 | NA3196 | NA3201 | | | |
| r | | | | 7.2 | " | " | | | |
| β | | | | .31 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3236 | NA3226 | NA3231 | | | |
| r | 7.2 | " | " | " | " | " | " | " | " |
| β | .44 | " | " | " | " | " | " | " | " |
| GAP | .070 | " | " | .080 | " | " | .090 | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA3056 | NA3046 | NA3051 | NA3011 | NA3001 | NA3006 | NA3101 | NA3091 | NA3096 |
| r | | | | 7.2 | " | " | | | |
| β | | | | .6 0 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3146 | NA3136 | NA3141 | | | |
| r | | | | 7.2 | " | " | | | |
| β | | | | .91 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3191 | NA3181 | NA3186 | | | |
| r | | | | 8.0 | " | " | | | |
| β | | | | .36 | " | " | | | |
| GAP | | | | .080 | " " | | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | NA3266 | NA3256 | NA3261 | | | |
| r | 8.0 | " | " | " | " | " | " | " | " |
| β | .54 | " | " | " | " | " | " | " | " |
| GAP | .070 | " | " | .080 | " | " | .090 | " | " |
| RPM | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 | 1100 | 1300 | 1500 |
| FILE | NA3086 | NA3076 | NA3081 | NA3041 | NA3031 | NA3036 | NA3131 | NA3121 | NA3126 |
| r | | | | 8.0 | " | " | | | |
| β | | | | .74 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | KNOCK | NA3166 | NA3171 | | | |
| r | | | | 8.0 | " | " | | | |
| β | | | | 1.18 | " | " | | | |
| GAP | | | | .080 | " | " | | | |
| RPM | | | | 1100 | 1300 | 1500 | | | |
| FILE | | | | KNOCK | NA3211 | NA3216 | | | |

Mode 3 behavior

Performance of the modified CFR engine towards the limits predicted by quasi-equilibrium thermodynamic models is given in this mode. Fig. VII-24 and 25 typify what we call Heat Balanced Cycle operation. It is evident from Mode 1 and 2 behavior that proper separation of fuel and air is responsible for this dramatic shift in the engine characteristic. The engine was run overly rich (beyond stoichiometric) only to demonstrate the full range of operation possible. Considering that best power was at A/F = 16, normal operation would be from this point, with an ISFC = 0.375 to best economy at an ISFC = 0.350. Since the output is 30-40% greater than the OTTO engine at this compression ratio and RPM, one can run at air fuel ratios from 18-23 at essentially constant specific fuel consumption. The RQI shown in Fig. VII-26 would then remain above 4000 while the Otto remains below 40. It should be noted that Fig. VII-26 is a single composite of the four curves of Fig. VII-25.

Comparing the Heat Balanced engine characteristics of Fig. VII-24 in the regime from best power to best economy with that of the Otto, it is evident that the Heat Balanced characteristic is rotated 90° from that of the Otto and displaced below the Otto to lower specific fuel consumption. Results at other RPM and compression ratios were similar and documented in Ref. 9.

A brief comparison of relative performance between modes 1, 2, 3 and the Otto is now given over a limited range of Heat Balanced operation at 1500 RPM. It is evident each Heat Balanced run could have been extended to far richer and leaner operation from Fig. VII-24 and 25, but as pointed out initially, runs were limited generally to five points.

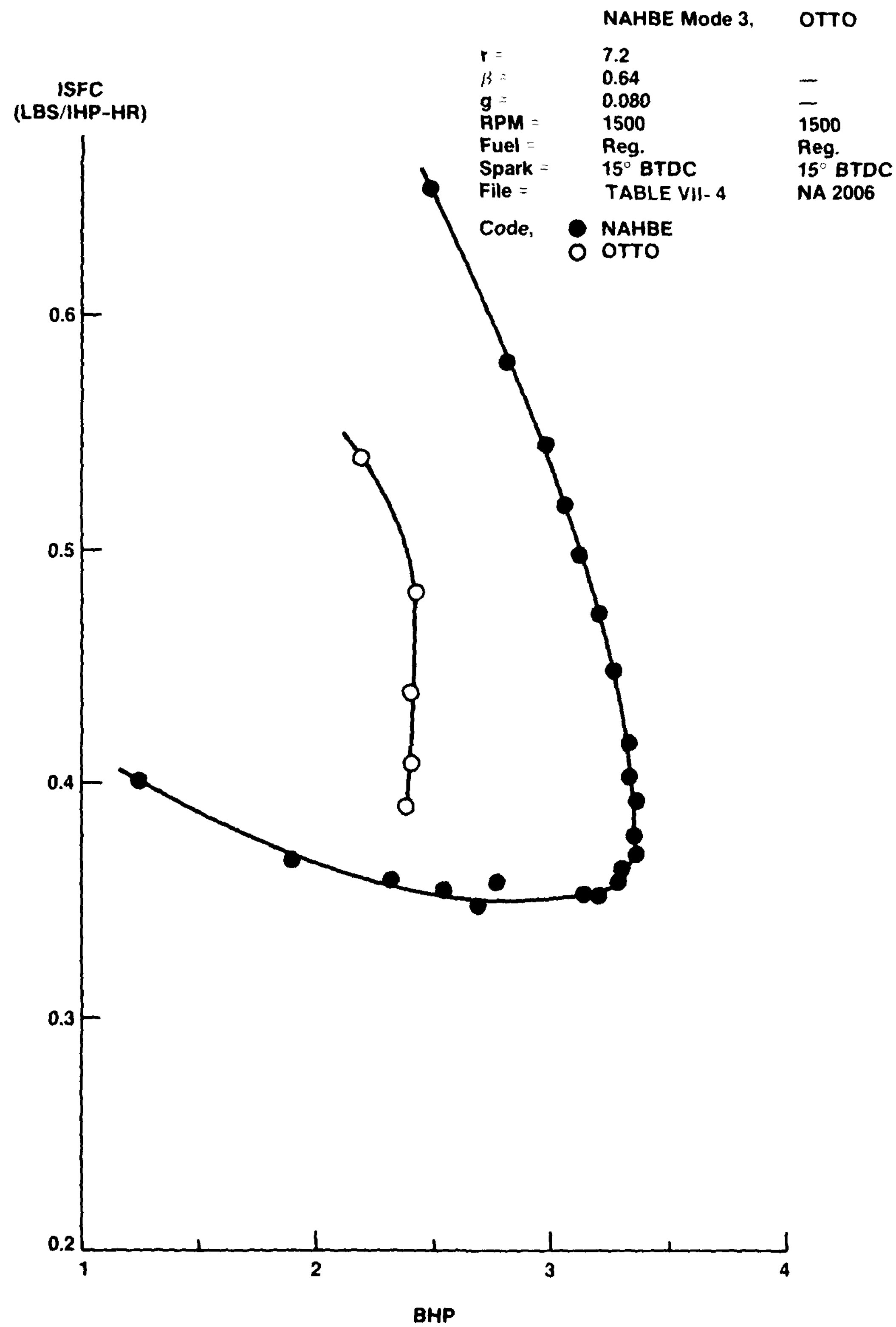



Fig. VII-24
OTTO, NAHBE — Mode 3 Comparison

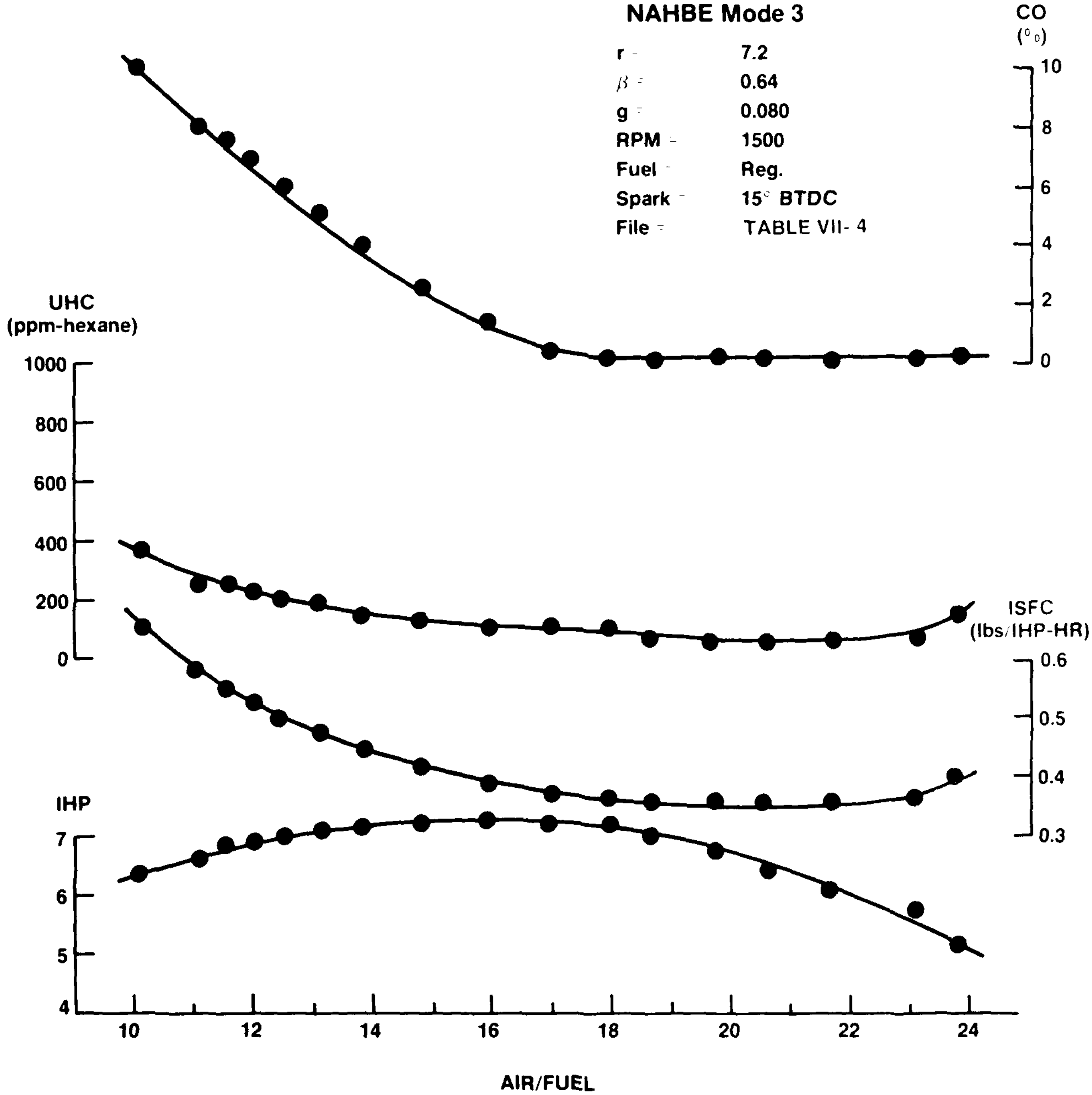



Fig. VII-25
NAHBE — Mode 3 Performance

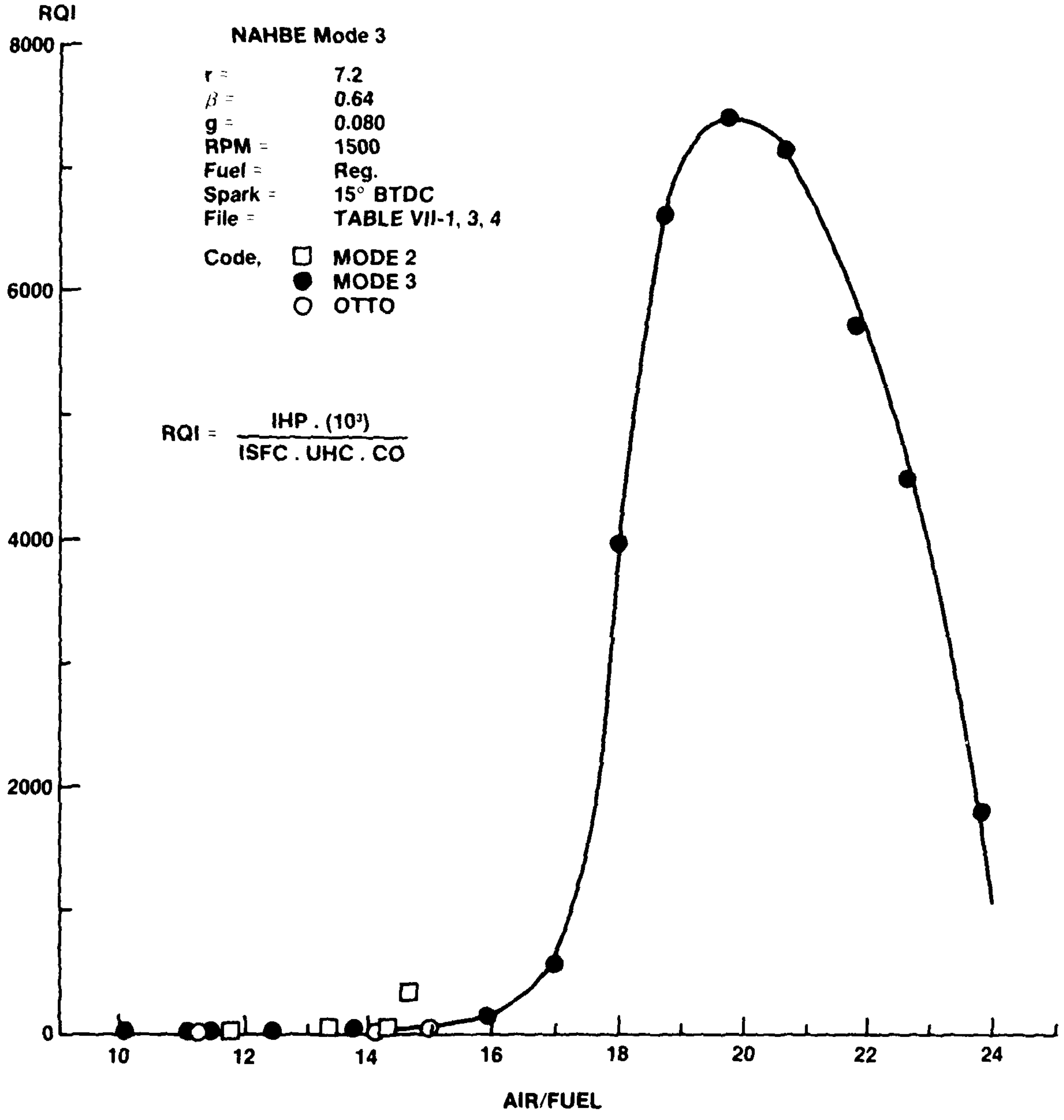



Fig. VII-26
NAHBE — Mode 3 Run Quality Index

TABLE VII-4

NAHBE OPTIMIZATION TEST SERIES, MODE 3
SUMMARY OF RUN CONDITIONS

FUEL TYPE 2 (MID RANGE)

| | | | |
|---|---|---|---|
| r | 5.6 | " | " |
| β | .41 | " | " |
| GAP | .080 | " | " |
| RPM | 1100 | 1300 | 1500 |
| FILE | OP110 | OP100 | OP105 |

| | | | |
|---|---|---|---|
| r | 6.6 | " | " |
| β | .57 | " | " |
| GAP | .080 | " | " |
| RPM | 1100 | 1300 | 1500 |
| FILE | OP125 | OP115 | OP120 |

| | | | |
|---|---|---|---|
| r | 7.2 | " | " |
| β | .64 | " | " |
| GAP | .080 | " | " |
| RPM | | 1300 | 1500 |
| FILE | | OP130 | OP135 |

| | |
|---|---|
| r | 7.2 |
| β | .64 |
| GAP | .080 |
| RPM | 1500 |
| FILE | OP140,OP145,OP150,OP155 |

The shift in output between modes is readily evident in Fig. VII-27 but a few runs in mode 2 did exceed Otto output by a few percent. With decreasing output from best power in Fig. VII-28, the emissions improved in all modes. Typical shifts in peak pressure between modes is given in Fig. VII-29 as well as exhaust gas temperature (EGT). The worst EGT behavior was experienced in Mode 2 with temperatures exceeding the leanest Otto values. In Mode 3 behavior, no EGT exceeded the leanest Otto value although the range was extended to A/F = 24.

The pressure difference ($\Delta p$) given in Fig. VII-30, is for the overall (primary and secondary) air flow metering orifice. Again the results are typical and show, since the flow rate is proportional to the square root, a 28% average increase in flow over the Otto. This seems to be the dominant factor leading to the increase in mean effective pressure as seen in Fig. VII-30. Mode 3 behavior of ($\Delta p$) is given in Fig. VII-31.

The last comparison we make is for Indicated Thermal Efficiency (ITE) in Fig. VII-32. The significant increase in efficiency is noted throughout Mode 3. In Fig. VII-26, the peak in RQI coincides with the peak in thermal efficiency at an air fuel ratio of 20. Although both plots extend over the same air-fuel range, it is not until emission levels improve that the RQI increases.

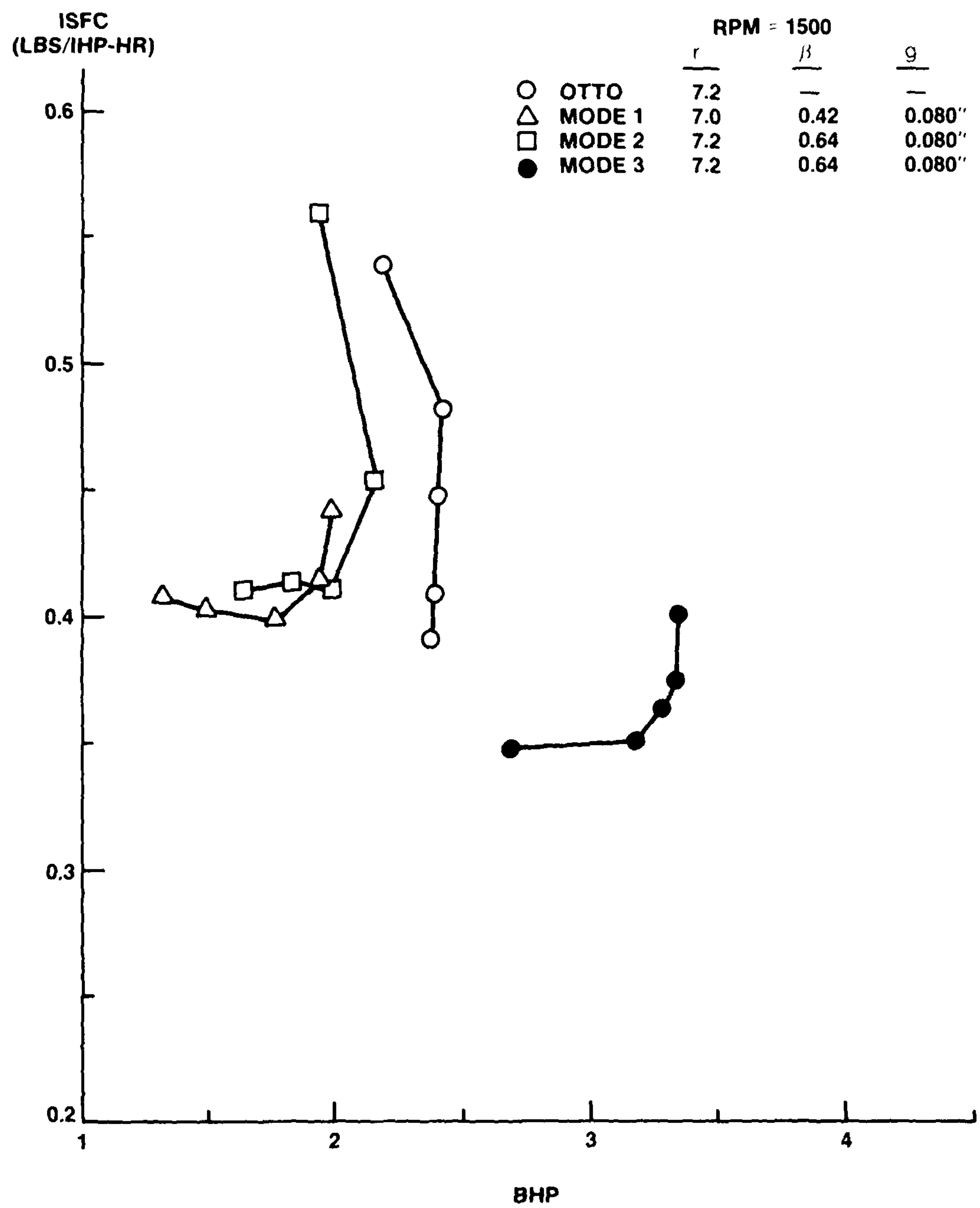



*Fig.* **VII-27**
**Comparison of all Modes with OTTO Behavior**

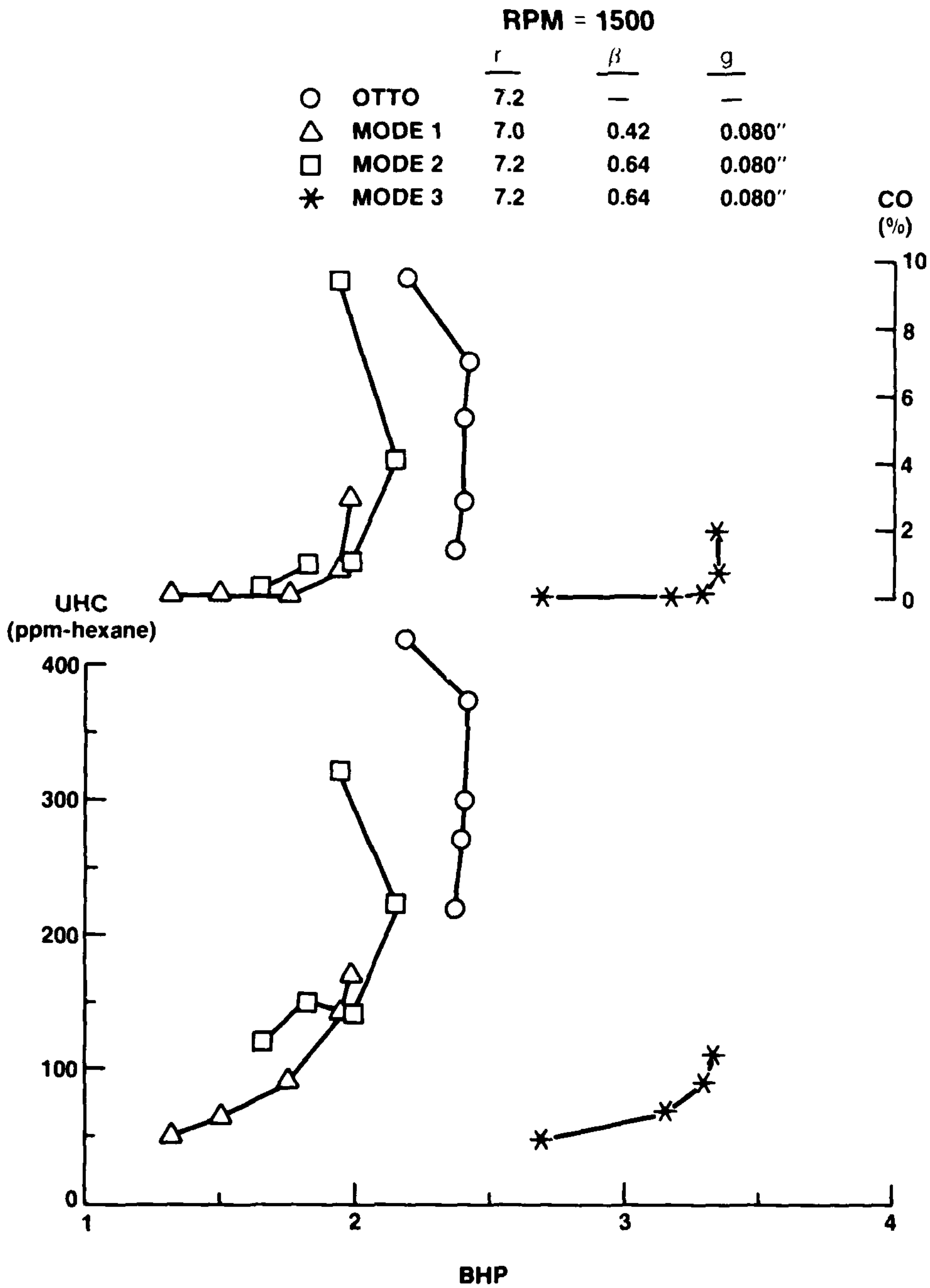



**Fig. VII-28**
**Comparison of Emission, all modes**

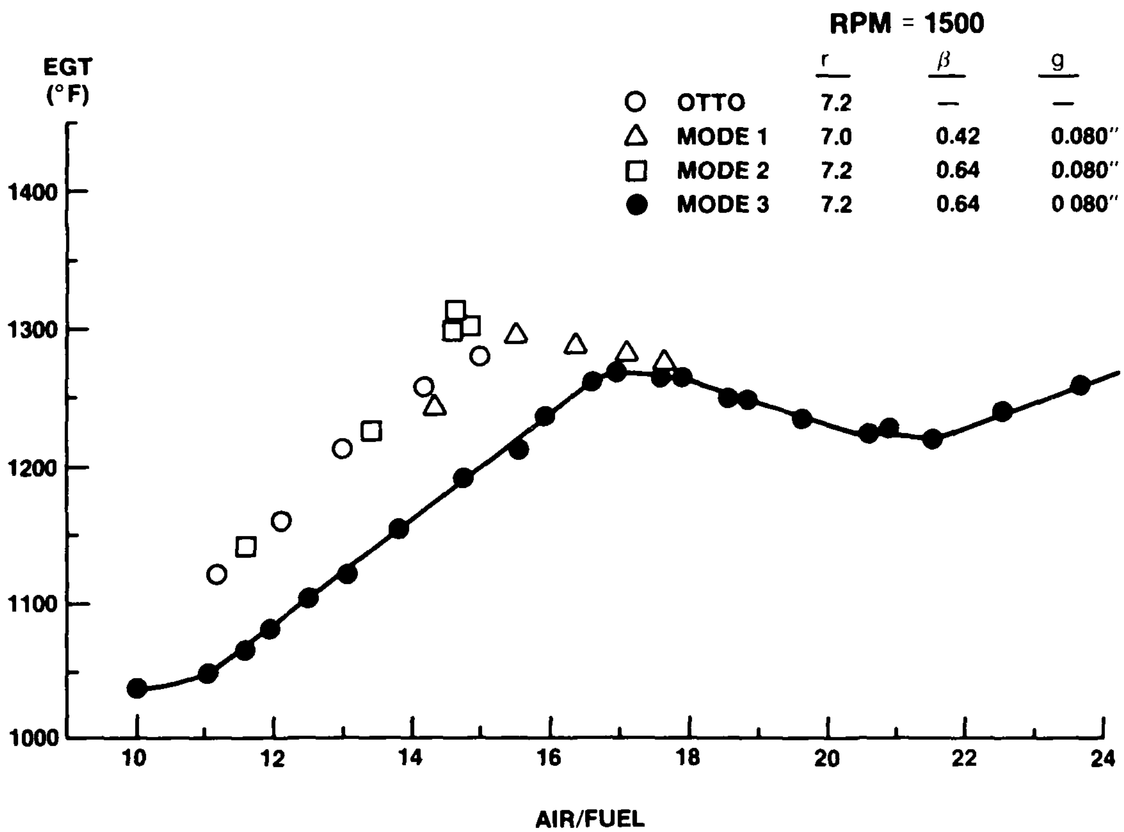



**Fig. VII-29**
**Exhaust Gas Temperature (EGT), All Modes**

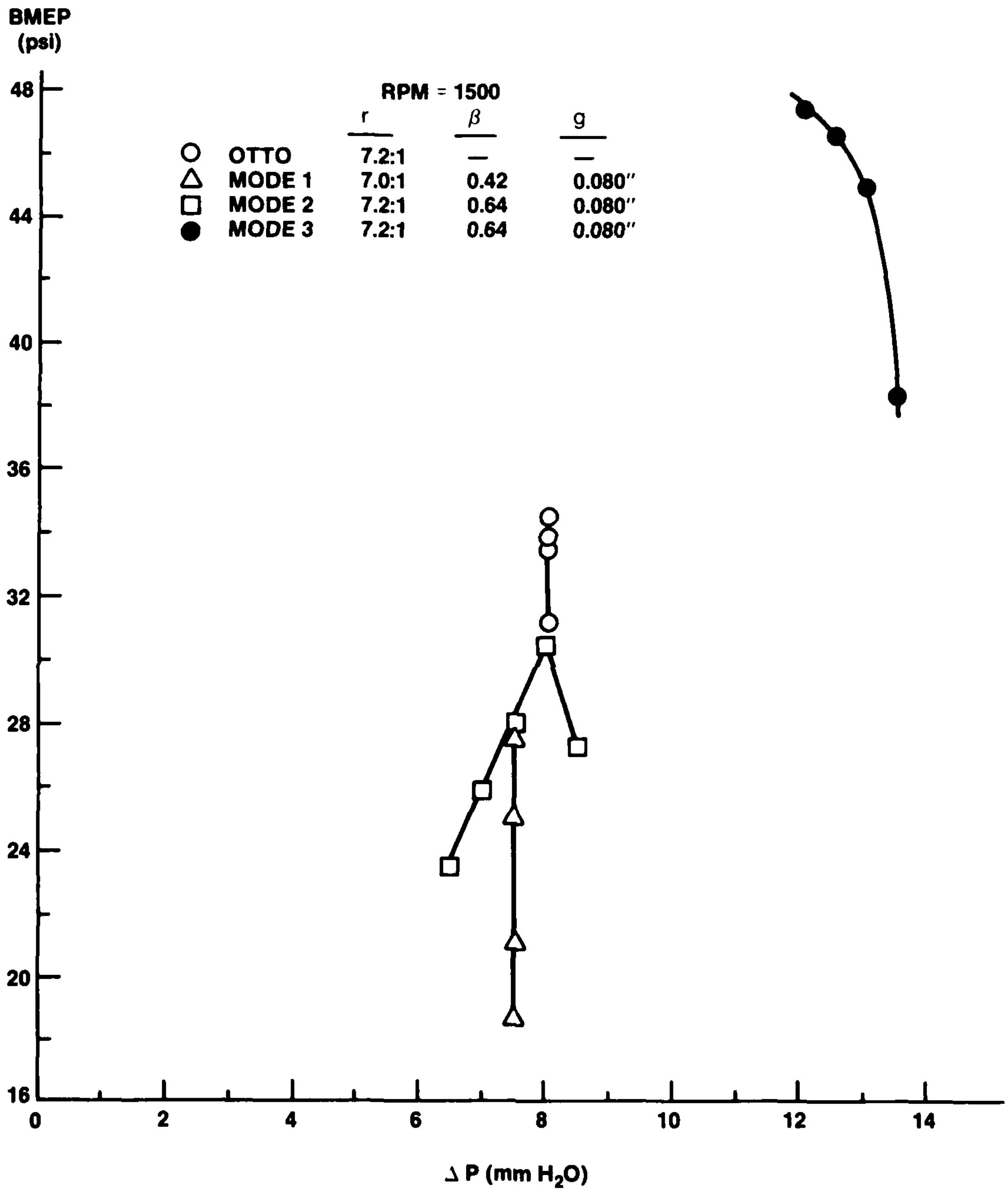



Fig. VII-30
Behavior of BMEP with Metering Orifice Pressure Drop,
All Modes

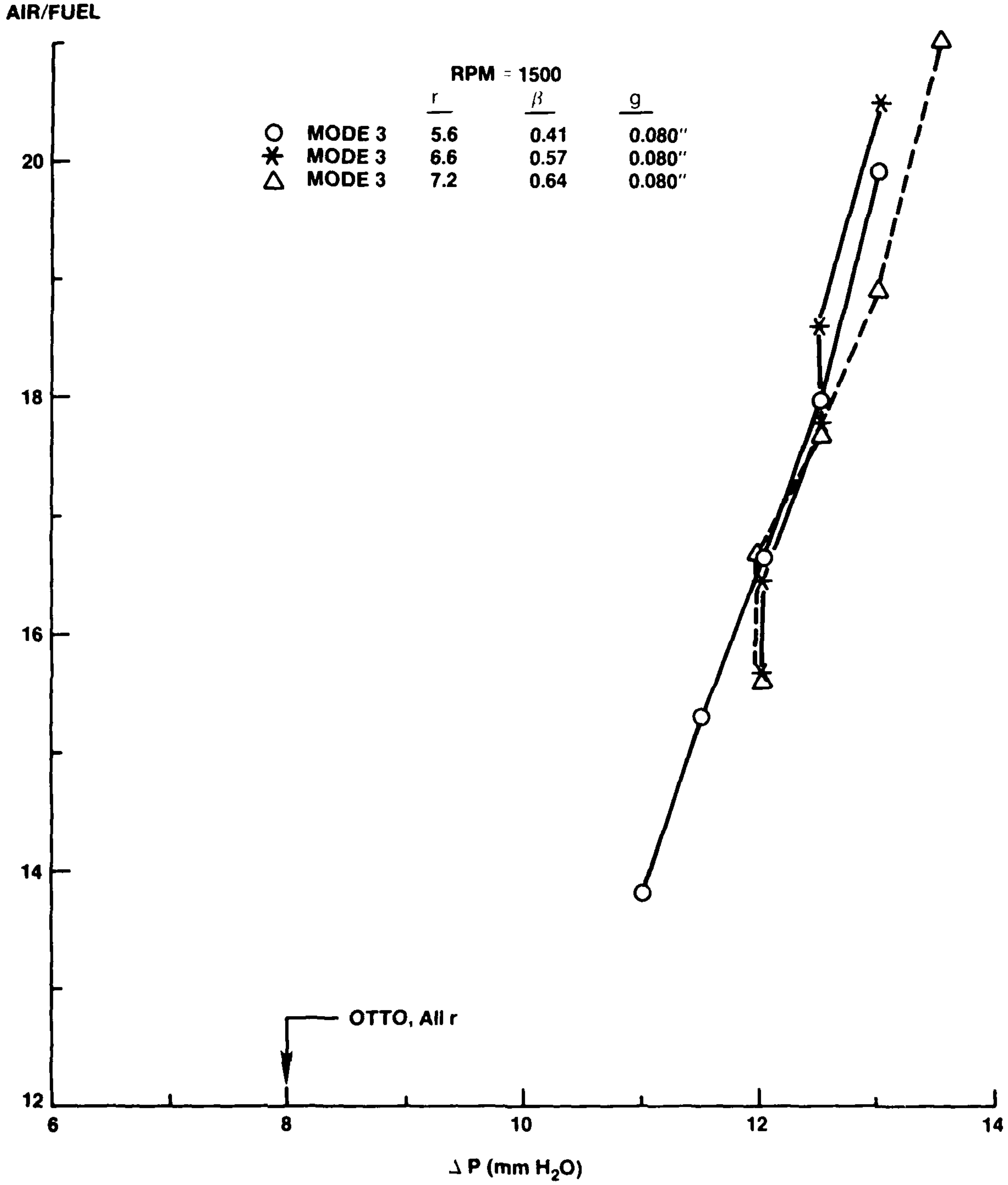



**Fig. VII-31**
**Behavior of Metering Orifice — Pressure Drop, Mode 3**

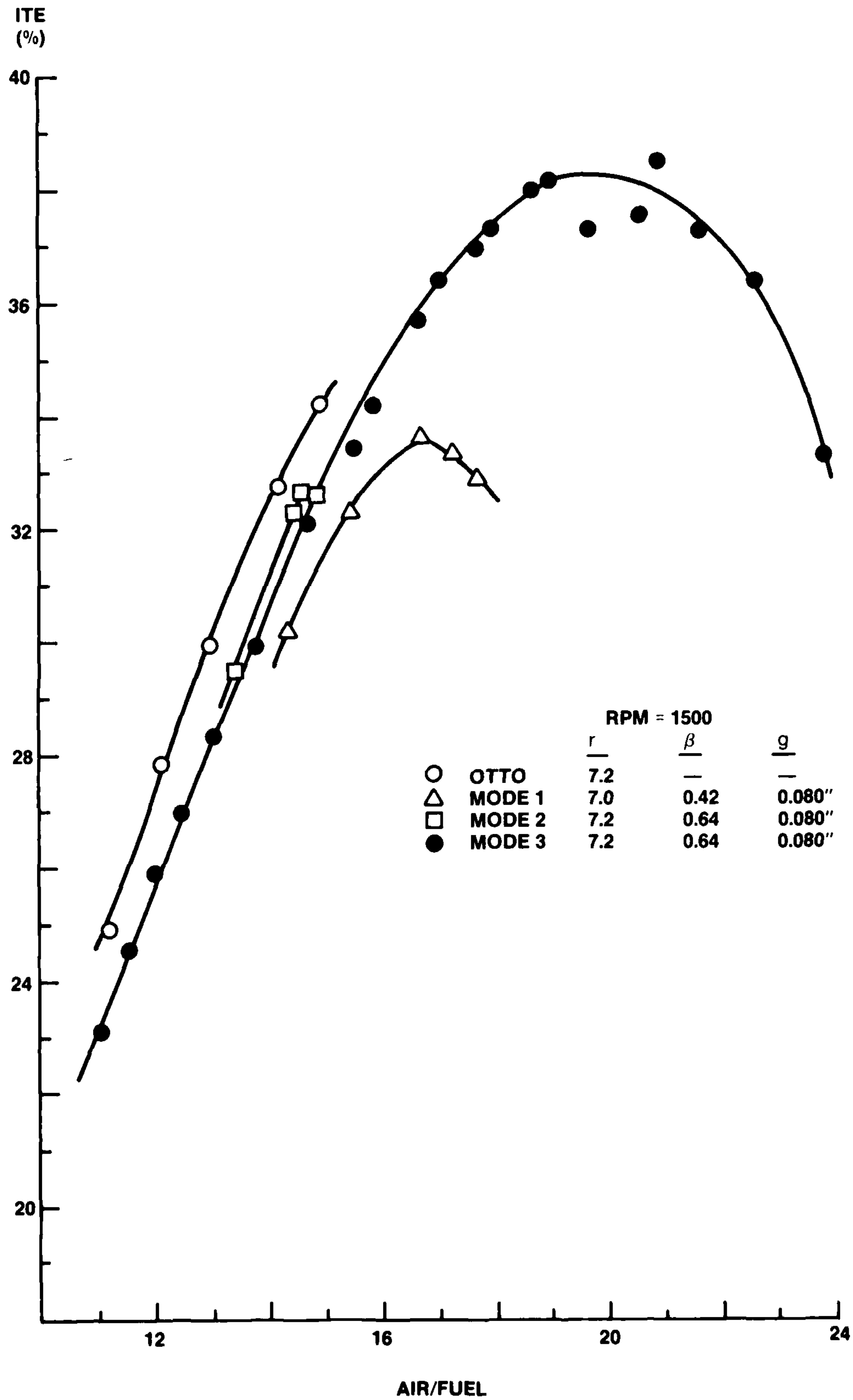



Fig. VII-32
Behavior of Indicated Thermal Efficiency, All Modes

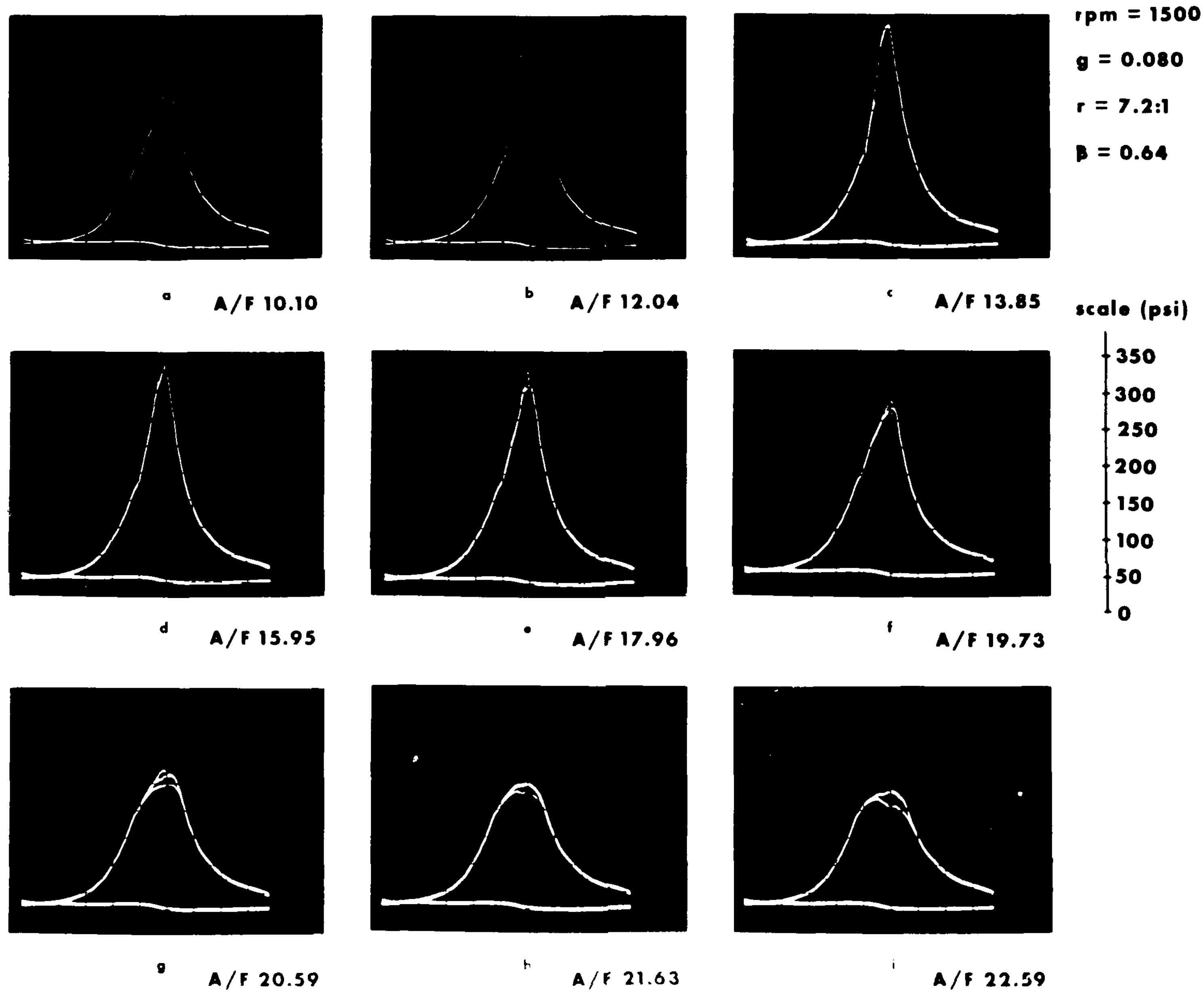



Fig. VII-33 Sample Scope Traces For Mode 3 Operation

# VIII. CONCLUSIONS AND RECOMMENDATIONS

A CFR engine configured to operate on the Heat Balanced Cycle (Mode 3) has successfully demonstrated its potential to outperform the conventional spark ignition CFR engine. Comparison of the two, run at a compression ratio of 7.2:1 and 1500 rpm, showed a average increase of 38.5% in BHP at the same ISFC, a 3.5% increase in peak ITE at best economy and significantly lower exhaust emissions. For the above condition there is a 10% improvement in ISFC at best economy and a 30% improvement at best power.

In achieving Heat Balanced operation, several aspects contributing to increased performance were identified. First, use of the pressure exchange cap optimized for balancing ratio and gap clearance improves engine performance as shown and extends the time for combustion as observed in a parallel study.[6] The prolonged time of combustion is a direct result of mass transport from the balancing chamber to the combustion chamber. Color Fastex photography, Fastex Schlieren and holographic interferometry[6] confirm the mass transport and extention of burn time. (The Heat Balanced method of separating a rich-lean charge should not be confused with a stratified charge engine for the pressure exchange cap allows, to a certain extent, gap quench to suppress ignition in the balancing chamber while the stratified charge engine relies on propagating of a flame from mixture of rich composition to lean.)

It is apparent from Fig. VII-24 that prolonged burn time extends reaction to the compact strongly bonded hydrocarbons that require additional time and energy to react. Exhaust emissions of a Heat Balanced engine operating in the preferred air/fuel region show that much of the unburnt hydrocarbons associated with the spark ignition engine exhaust is absent.

The amount of CO emitted during Heat Balanced operation is 1% or less at best power and 0.05% for most of the preferred operating region.

Unfortunately, this study did not sample $NO_x$ because of constraints on manpower and instrumentation but a trend can be predicted.* Figure VIII-1 presents the classical $NO_x$ vs Equivalence Ratio curve. Considering that Otto engine extracts best power at an equivalence ratio slightly greater than 1.0 and operates within a regime of air/fuel ratio of from 14 to 16:1 (or equivalence ratio 1.08 to 0.95) it can well be seen that the Otto engine produces high amounts of $NO_x$ at peak power and the percentage increases as the engine is leaned out. The Heat Balanced engine also extracts peak power in a high $NO_x$ range, but as it is leaned out to a preferred operating region, the $NO_x$ continually decreases while equivalence ratio approaches approximately 0.8. Furthermore, combustion is initiated in a rich mixture with additional $O_2$ added in time as a result of wave interaction. Both features favor reduction in $NO_x$. Therefore, one has control over the level considered acceptable and the engine can be designed to a particular standard.

It is known that the gain in output in some high performance engine is directly related to increased manifold pressure and the attendant increase in mass of air-fuel per cycle. Some of the Heat Balanced increase in output is directly related to this same effect. This can best be verified by comparing the amount of air taken in by both the Otto and Heat Balanced engines, i.e., by examining the intake airflow metering orifice pressure drop. The Otto and modified engine in Mode 1 operate with constant intake airflow ($\Delta p$) for constant rpm and varying air/fuel

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*Unpublished earlier experiment of work in a limited regime indicated significant reductions in $NO_x$

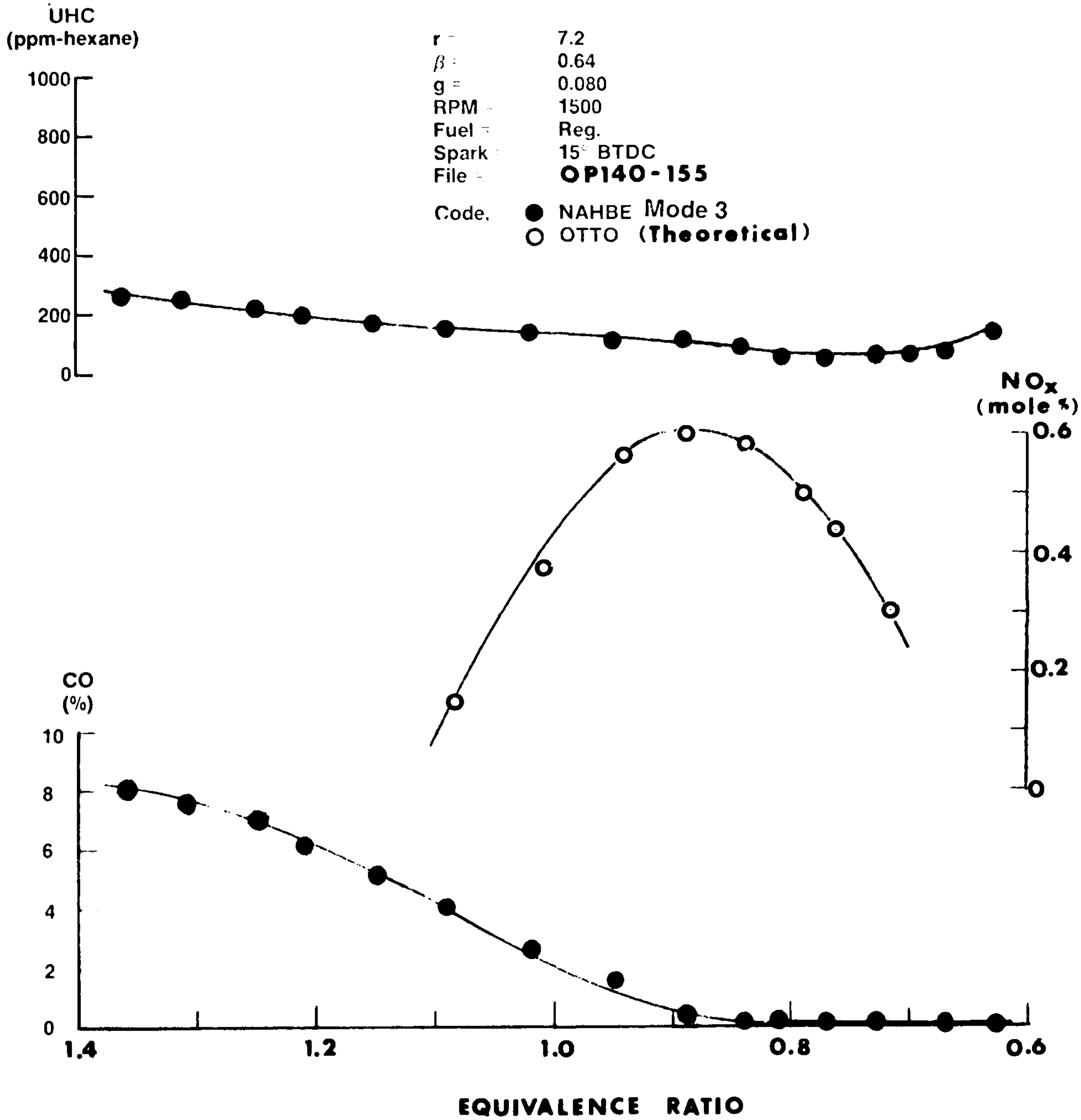


**Fig. VIII-1**

**Standard Otto $NO_x$ Projection**

ratio (Fig. VII-30). Mode 2 displays variable ($\Delta p$) for the same conditions (Fig VII-30) whereas Mode 3 demonstrated a 28% increase in overall intake air as seen in the increased ($\Delta p$) as the air/fuel ratio increased at fixed rpm (Fig VII-30).

Mode 3 operation is considered the preferred method of capturing the Heat Balanced cycle. It is recommended that any CFR follow-on work use an edge gap clearance of .080", a $\beta$ of 0.080 at a compression ratio of 8.0:1 and Mode 3 carburetion.

The Run Quality Index (RQI) is presented to help identify and optimize Heat Balance Cycle operation. Neither the Otto nor Mode 1 or 2 results caused as significant a rise in the RQI as in Fig. VII-26, with Otto results generally well below 100 and some Mode 2 results approaching 2000. Only Mode 3 performance within the preferred limits of air/fuel ratio leads to increases on the order of those shown in Fig. VII-24. The RQI appears to be more useful than thermal efficiency since it is sensitive to emissions as well as consumption.

One could summarize the conclusion of each NAHBE mode of operation with respect to Otto operation as follows:

Mode 1: Fixed geometry secondary and primary air

a. A shift in Air/Fuel ratio to leaner operation occurs.
b. Emissions are generally improved.
c. Maximum output is lower.
d. Specific fuel consumption is relatively constant from best power to best economy with best power ISFC lower and best economy ISFC higher.

e. A wider range of output at each RPM is possible.

f. Peak pressures are lower.

g. Peak exhaust gas temperatures are equal or slightly higher

h. More secondary air is required since best power occurs at too rich an air/fuel ratio.

Mode 2: Fixed geometry secondary and variable primary air.

a. A/F see Mode 1, a.

b. Emissions, see Mode 1, b.

c. Maximum output is higher in a few cases, but generally lower.

d. Wider variation in specific fuel consumption; using choke plate is of little advantage.

e. Output range, see Mode 1, e.

f. Peak pressures and peak output are related to $\beta$, though complete behavior will not be understood until related to Mode 3 behavior. Best $\beta \cong 0.80$ at $r = 8.0$ and gap = 0.080 inches.

g. Peak E.G.T, see Mode 1.g.

h. See also Mode 1. h.

Mode 3: Variable geometry secondary and fixed geometry primary air

a. Greatest variation occurs in A/F ratio, stable operation from A/F = 10 to 22.

b. Emissions and output are best at A/F = 20, according to RQI (CO < 0.1%, UHC < 100 ppm), also best thermal efficiency at A/F = 20.

c. Maximum output is considerably greater, at 1500 rpm and $r = 7.2$, average increase is 38.5%, results are

similar at all RPM.

d. Specific fuel consumption is about 10% lower at best economy and 30% at best power, engine can be operated at nearly constant ISFC at any rpm over wide load range.

e. Wider range of output than other mode.

f. Peak pressures are generally higher.

g. Peak exhaust gas temperatures are lower and DECREASE with increasing A/F ratio to A/F $\cong$ 22, than increase.

h. Separation of air/fuel charge is vital to achievement of this cycle. Cycle parameters, pressure, temperature, output, can be controlled by proper air/fuel system as well as optimization of internal engine geometry. The RQI is useful for comparative studies as well as optimization of this engine.

Follow-on work is recommended to identify the full potential and any weakness of the Heat Balanced concept. The engine's multi-fuel capabilities in both the spark ignition and compression ignition modes are of immediate interest. Unpublished results using a number of fuels, among them, propane, alcohol, gasohol, diesel oil, and mixtures of some of these demonstrate the engines multi-fuel potential. Studies in fuel delivery systems for this engine are also needed.

Finally, a detailed materials study is needed. Heat Transfer associated with the pressure exchange cap is complex and usually testing of several designs is required before a working model is achieved. Better understanding of time dependent heat transfer - material interactions would greatly simplify the design effort.

## IX. REFERENCES


1. Pouring, A. A., Blaser, R. F., Keating, E. L., and Rankin, B. H., The Influence of Combustion with Pressure Exchange on the Performance of Heat Balanced Internal Combustion Engines, S.A.E. Paper 770120, February, 1977.

2. Adams, J. A., Heat Transfer Analysis of the NAHBE Piston Cap, USNA Engineering and Weapons Report EW-11-77, September, 19[illegible].

3. Adams, J. A., Heat Transfer Analysis of the NAHBE Piston Cap, Part II, An Extension USNA Engineering and Weapons Report EW-[illegible], February 1979.

4. Hasson, D. F., Materials Analysis for Selection of Alloys for the NAHBE Pressure Exchange Cap, USNA, Engineering and Weapons Report EW-3-79, May, 1979.

5. Pandalai, R. P., Preliminary Investigation of the Nonsteady Combustion and Flow Processes of the Naval Academy Heat Balanced Engine (NAHBE), USNA, Engineering and Weapons Report EW-10-78, May, 1978.

6. Pouring, A. A. and Rankin, B. H., Time Dependent Analytical and Optical Studies of Heat Balanced INternal Combustion Engine Flow Fields, USNA Engineering and Weapons Report.

7. Keating, E. L., Blaser, R. F., and Pouring, A. A., Quasi-Equilibrium Air Standard Heat Balanced Cycle Analysis, SAE paper #789036, August, 1978.

8. Keating, E. L. and Pouring, A. A., Quasi-Equilibrium Fuel-Air Heat Balanced Cycle Analysis, USNA Engineering and Weapons Report EW-14-79.

9. Failla, C. C. and Pouring, A. A., CFR - OTTO - NAHBE Parametric tests; Data Calculated Results, USNA Engineering and Weapons Report EW-15-79.

[illegible], and Taylor, E. S., The Internal Combustion Engine, [illegible] Textbook Co[illegible], Scranton, Pennsylvania, 1961.

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